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THE PANAMA CANAL

SOME INTERESTING MECHANICAL FEATURES

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THE Panama Canal is a rather difficult subject to treat, because there are so many wonderful features that it is hard to know where to begin. We have been asked a number of times: "What is the most impressive feature of the Canal?" Well, that depends. One man, who lives in the mosquito belt of New Jersey, contended, after visiting Panama, that the construction of the canal was not comparable with the work of extermination wrought by the sanitary engineers. Another pilgrim to the isthmus seemed to almost overlook the canal in his enthusiasm for the "large, airy and comfortable" Y. M. C. A. buildings of the Canal Zone. Then there was the civil engineer who exhausted his stock of adjectives in praise of the dam at Gatun, and no doubt a mechanical or electrical engineer would consider the locks the greatest feature. So the "impressive features" are almost as numerous as the different classes of people who see the canal. Surely, the locks with their ingenious operating and controlling mechanism are not far from first place; but, whatever the verdict, we shall reluctantly pass by the dams and cuts, owing to the limited space, and confine our attention principally to describing some of the more interesting mechanical features. In order to show clearly the relation between the various details that will be referred to, a brief description of the canal will be given first, even at the risk of repeating what may be known to many readers of MACHINERY. Perhaps the general arrangement and operation can best be understood by taking an imaginary trip through the canal, assuming that it has been completed and is ready for business.

A Trip through the Canal

In passing from the Caribbean Sea to the Pacific, a ship first enters a sea level channel which extends for a distance of 7 miles to the foot of the locks at Gatun. Here the ship passes through a series or flight of three twin locks (see Fig. 1) and is lifted 85 feet to the level of Gatun Lake. It first enters the forebay of the locks and ties up to a central wall which extends some distance beyond the side walls, to serve as a wharf or mooring wall. The vessel waits here until the electric locomotives, which are to be used for towing all ships through the locks, come down and make fast their hawsers, after which it is towed into the lower lock chamber. The gates are then closed and enough water admitted to raise the vessel one third the total height (or 28⅓ feet) and to the level of the water in the middle lock which is then at low level. The vessel next passes into the middle lock where it is raised to the low level of the upper lock, which it finally enters and is elevated to the level of Gatun Lake. As is generally known, Gatun Lake was formed by building the large Gatun dam, thus impounding the waters of the Chagres River and its tributaries. (See map of the Canal Zone, Fig. 2.) When passing through this lake, the ship may run full speed for a distance of approximately 24 miles or as far as Culebra cut. The length of Culebra cut, which is really an arm of the lake, is nine miles, and after passing it at reduced speed, the vessel enters a lock at Pedro Miguel and is lowered 30⅓ feet (from mean tide) to the level of a small artificial lake. This lake, which has an elevation of 54⅔ feet, is then transversed for a distance of 1½ mile to Miraflores. Here the ship enters two locks in a series, is lowered to sea level, and passes out to the Pacific through a channel about eight miles long. The length of the canal from shore-line to shore-line is about 40 miles, and the entire length from deep water in the Caribbean to deep water in the Pacific, is about 50 miles. The estimated time required for the passage of a medium-sized ship through the entire canal, is from 9½ to 10 hours, and for large vessels, from 10½ to 11 hours, the time depending somewhat upon the size of the ship and its speed. The general direction of the canal is from northwest to southeast, as shown by the map,

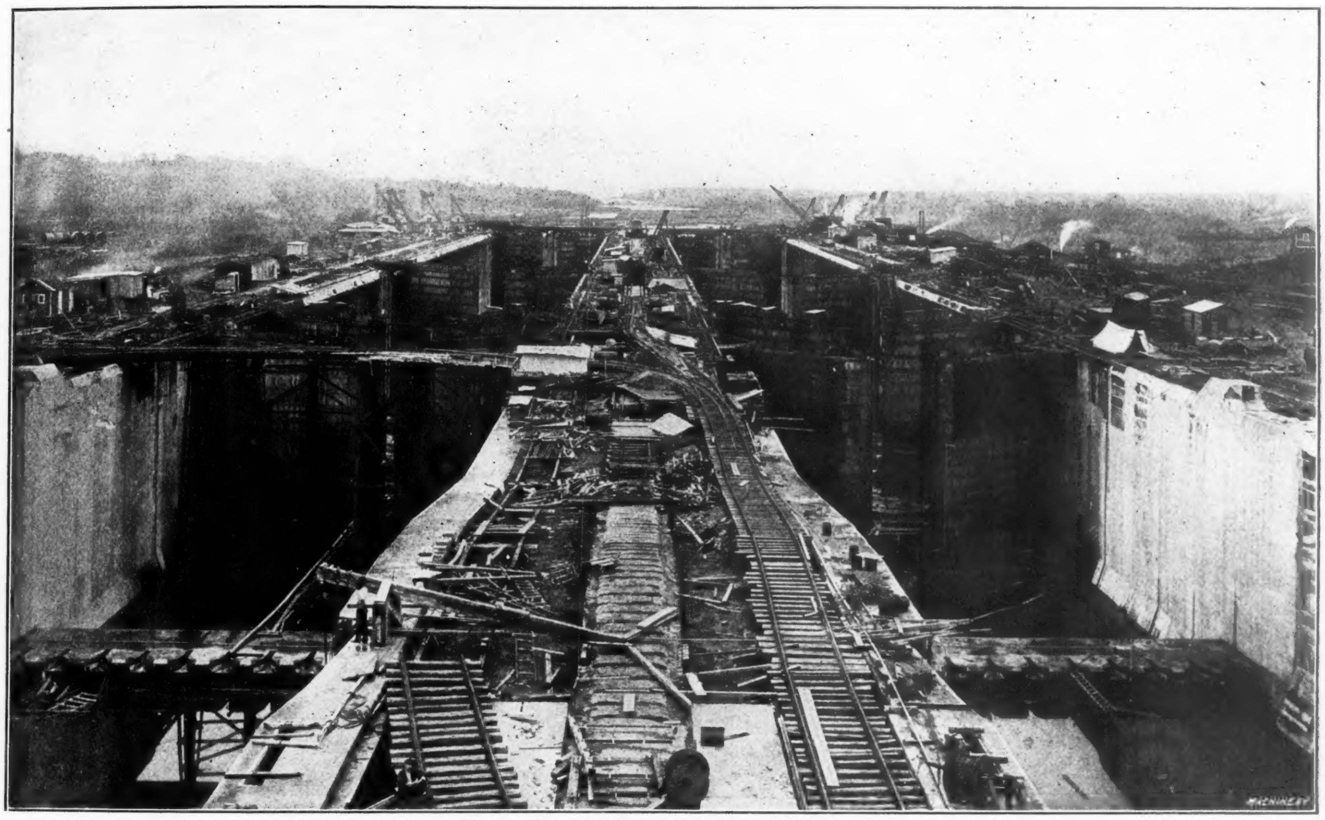

Fig. 1. General View of Upper Locks at Gatun

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and not from east to west, as is often supposed. The Pacific entrance near the city of Panama, is about 22½ miles *east* of the Atlantic entrance near Colon.

Arrangement of the Locks

All of the locks are arranged in pairs (as shown by the general view, Fig. 1) so that vessels can pass through in opposite directions at the same time, one side being used for ships bound for the Pacific and the other side for those going towards the Atlantic. The general construction of these twin floor, thus reducing the disturbance when filling or emptying the lock. The main feed culverts are 18 feet in diameter and have an area of 254 square feet, which is about the area of the Hudson River tunnels of the Pennsylvania R. R.

The flow of water in and out of the main culverts is controlled by large gate valves located near the gates at the ends of each lock chamber. The center wall culvert *D* is illustrated in Fig. 5, which also shows the chamber for one of the gate valves. The lateral culverts leading from the central wall are all controlled independently by cylindrical valves at *E*,

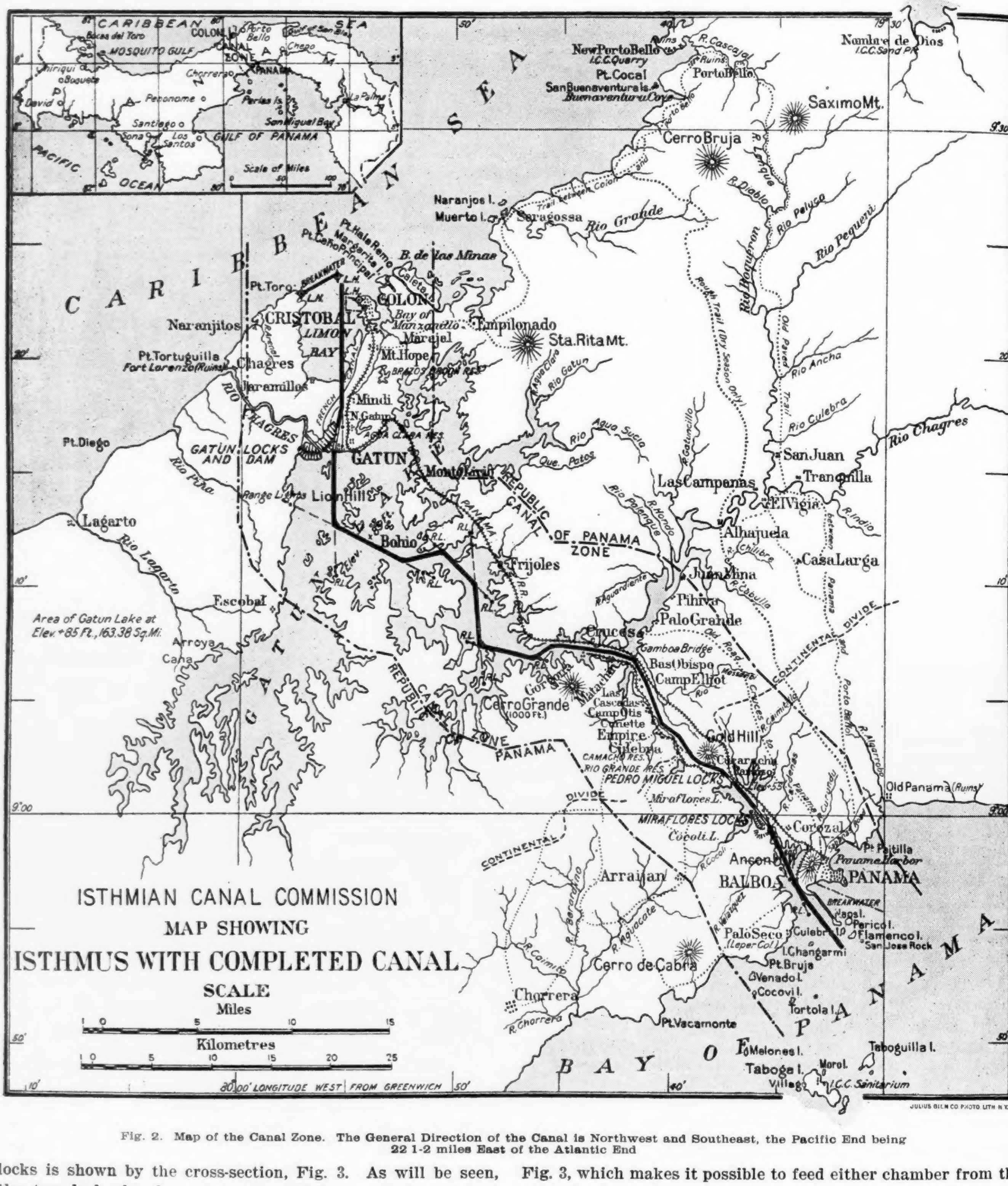


Fig. 2. Map of the Canal Zone. The General Direction of the Canal is Northwest and Southeast, the Pacific End being 22 1-2 miles East of the Atlantic End

locks is shown by the cross-section, Fig. 3. As will be seen, the two lock chambers are separated by a massive central wall. These chambers will be filled and emptied through large culverts located in the side walls at *G* and in the center wall at *D*. These main culverts extend the full length of the lock and connect with the chambers by means of lateral culverts *H* having circular openings *F* through the floor of the lock. The lateral culverts from the center wall alternate with those from the side walls. This arrangement is for distributing the water evenly over the entire area of the lock Fig. 3, which makes it possible to feed either chamber from the central culvert. For example, to feed to the right, all the cylindrical valves on the left side would be closed, whereas to feed to the chamber on the left, the valves on the opposite side would be closed. This independent control also makes it possible to transfer the water from one side to the other, by simply opening the cylindrical valves on both sides, which will be done to effect a saving of water, as will be explained later.

In order to fill a lock for elevating a ship, the gate valves at

the lower end of the main culverts are closed and those at the upper end are opened. The water then flows from the upper level into the main and lateral culverts and up into the chamber through the holes in the floor, the flow continuing until the water level coincides with that of the lock above. To empty the lock, the order of operation is simply reversed, the valves at the upper end being closed and those at the lower end being opened. The water then flows out through the lateral and main culverts into the lower pool or level. The average time required for filling and emptying a lock chamber will be about fifteen minutes, and the time required for passing a vessel through the six locks of the canal is estimated at three hours. Of this time, 1½ hour will be required at the Gatun and about the same time for the three locks on the Pacific side. It is estimated that the minimum interval between consecutive ships at any lock, will be one hour, giving a maximum of 24 lockages per day, in each direction.

The amount of water that can be saved by transferring it from the chambers of one flight to those on the opposite side, will depend somewhat upon circumstances. It is not probable that a complete saving can be effected, as this would require that at each lockage, one-half the water in the upper lock of a flight be transferred to its twin chamber. This, however, will be possible at the end of every up lockage, when the locks in the downward flight are left empty after use, and those in the upward flight are left full, for when an up-bound ship leaves the upper lock, one-half the water can be transferred to the twin chamber of the downward flight. The up-bound lock would then be left at low level for the next ship, and the downward lock would be left at high level for lowering a ship. While this economy in the use of water is possible, there will be no danger of an inadequate water supply. During eight or nine months of the year, Gatun Lake will be kept full by prevailing rains, so that it will only be necessary to store a surplus for the dry season, which lasts only three or four months. During the rainy season, the lake will be kept at an elevation of 87 feet above mean sea level, at which time the minimum channel depth will be 47 feet. As 40 feet is ample for navigation, there would be a surplus of 7 feet for the dry season. After making allowances for evaporation, leakage and use of water at the hydro-electric power plant, this surplus is greater than would be required, even though the maximum number of ships passed through the canal every day. The normal area of Gatun will be 164 square miles.

Some idea of the size of the locks can be obtained from the following dimensions: Each chamber has a usable length of 1000 feet and a width of 110 feet. The side walls are about 50 feet wide at the floor and from a point 24½ feet above the floor, they are stepped inward to a width of 8 feet at the top. The central wall is 60 feet wide and about 81 feet high. As the sectional view, Fig. 3, shows, this central wall is divided into passages or galleries at the top. The upper passageway *A* is to give access to the valve- and gate-operating mechanisms, which are installed in this central wall. The middle gallery *B* contains the electric wires for supplying power to the various motors, and the lower gallery *C* is for drainage.

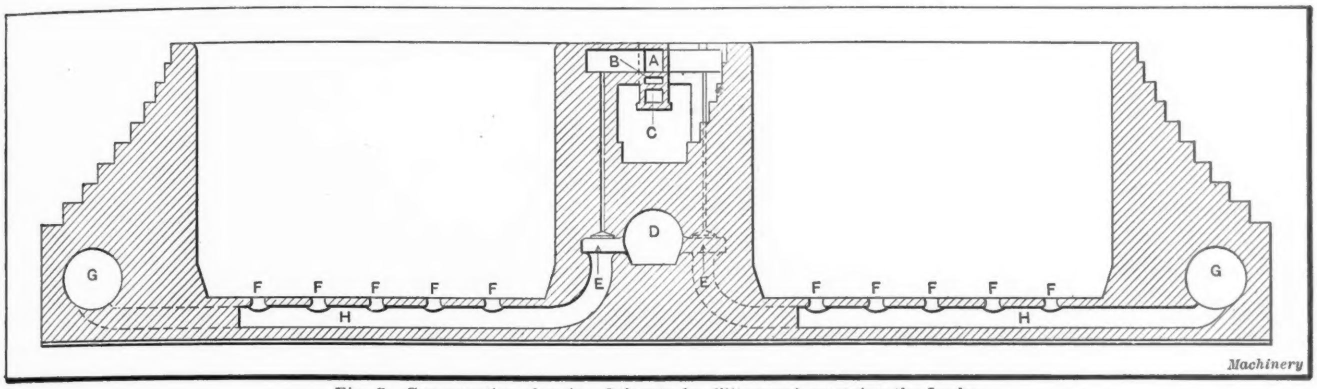


Fig. 3. Cross-section showing Culverts for filling and emptying the Locks

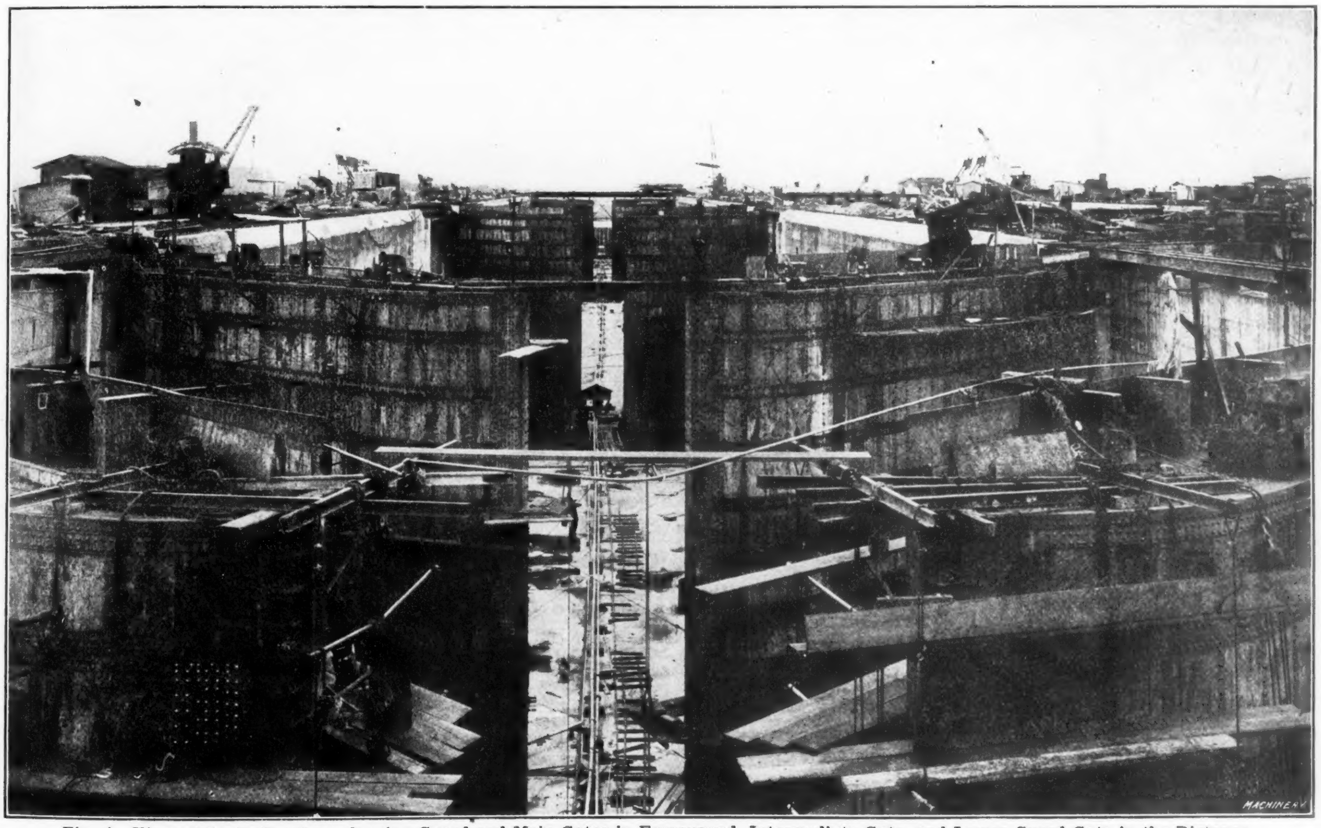


Fig. 4. View of Lock Chamber, showing Guard and Main Gates in Foreground, Intermediate Gate, and Lower Guard Gate in the Distance

The Lock Gates

The lock gates are built-up steel structures which have a width of 65 feet, a thickness of 7 feet, and a height varying from 47 to 82 feet. The upper chamber of a flight and the outer end of the lower chamber, are equipped with two sets of gates instead of a single pair, an auxiliary guard gate having been installed to protect the main gate from being rammed, in case a vessel does not stop in time. In addition to the gates at each end, there are intermediate gates which divide the 1000-foot locks into chambers 600 and 400 feet long, in order to save time and economize in the use of water when ships are passing through that are not large enough to require the entire lock. Fig. 4, which is a view of one of the upper chambers at Gatun, shows the guard and main gates in the foreground, the intermediate gate 400 feet away, and the lower guard gate in the distance. Each of the half gates or "leaves," as they are called, weighs from 390 to 730 tons, and

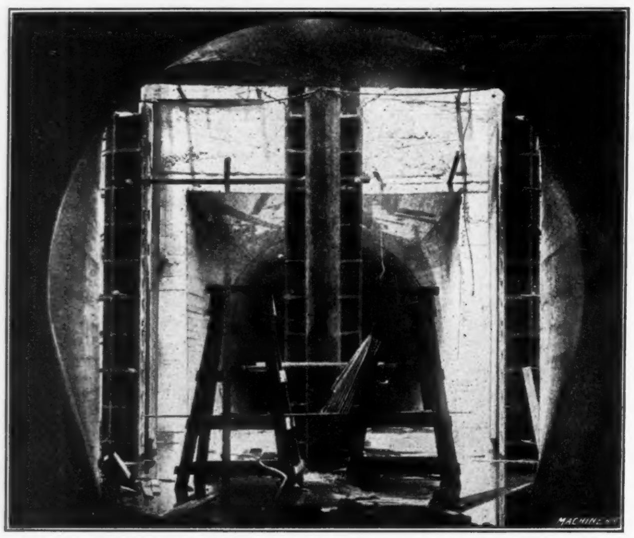


Fig. 5. Center Wall Feed Culvert, showing Chamber for Gate Valve

most of this weight is supported by an air chamber formed in the lower half and sub-divided by three water-tight bulkheads. By supporting the gates in this way, the bearings upon which they swing, are relieved of excessive stresses. The upper part of the gate is arranged with openings on the up-stream side, through which the water can flow in or out of the gate at the same time that it enters or recedes from the lock. With this arrangement, the weight of the gate is automatically increased as the water rises to a higher level, thus counteracting the increased lifting effect as the air chamber is submerged to a greater depth. Each gate swings on a hemispherical pintle which is secured to the concrete floor near the wall, and connection is made with an anchorage at the top by a vertical pin which forms a hinge.

Gate Operating and Locking Mechanism

The mechanism for opening and closing these massive gates is shown in Figs. 6 and 7, the latter being a diagrammatical plan view. The power is obtained from a 50 H. P. motor *A* which drives through spur and bevel gears, a pinion meshing with the large crank-gear *B*, which is connected to the gate leaf by a rod or pitman *C*. The gear *B* rotates through an arc of 197 degrees, when swinging the gate from the open to the closed position, or *vice versa*. When the gate is open or closed, the crank is approximately on the dead center; hence, a powerful but slow motion is obtained at both extremes of the gate's travel, which is desirable because the resistance to motion is greatest at these points. This increased resistance is caused by the water which, at the point of opening or closing, has a smaller space through which to escape around the edges of the leaf, and therefore tends to pile up and retard the movement of the gate. The connecting-rod or pitman *C*, instead of being rigid, has a spring buffer or shock absorber at *D* that gives a certain amount of flexibility, which is advantageous owing to the inertia of the gates, as it tends to eliminate shock and jar on the mechanism when starting. Moreover, when one leaf comes against the other, the fine adjustment that

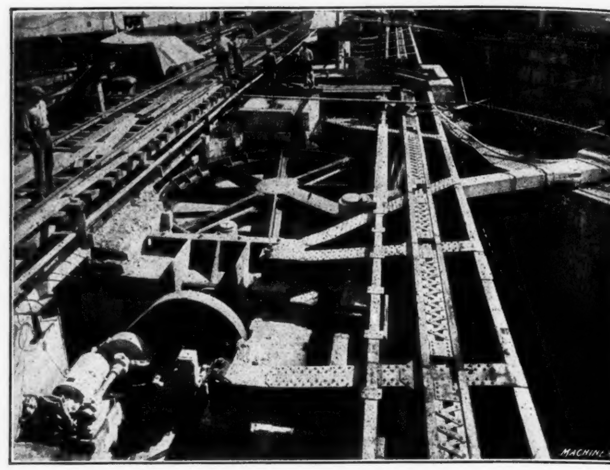


Fig. 6. Mechanism for opening and closing a Lock Gate

would be necessary with a rigid connection, is obviated by having a pitman that will yield slightly. When the gates are opened, they enter pockets or depressions formed in the walls. During a test of one of the gate-moving machines, a gate was swung "in the dry" in one minute and 48 seconds, which is 12 seconds less than the estimated time.

In Fig. 8 is shown a locking device which has been designed to hold the gates securely against wave action and, at the same time, force the gates in order to form a perfect miter or joint between the two leaves. A crosshead is mounted at the top of one of the miter posts and it is moved horizontally by screw *B*, which, in turn, is rotated by motor *A* through gears *D*. As the crosshead moves outward, two jaws *E* which are connected to the crosshead by toggle links, are forced to close around a pin, which is so located on the opposite gate that when it is firmly held by jaws *E*, the two gate-sections are

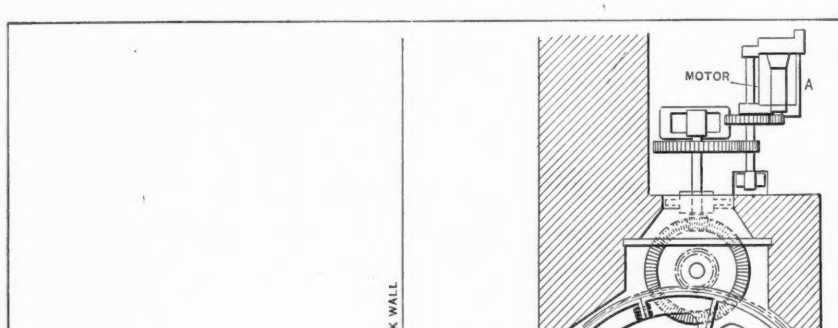


Fig. 7. Plan View of Gate Operating Mechanism

properly joined. This is a new device and is to be carefully tested before being extensively applied.

The Protective Chains for the Gates

In addition to the auxiliary or guard gates previously referred to, the locks are equipped with large protective fender

chains which are stretched across the chambers some distance in front of the gates. In case a vessel entering a lock should continue to move in toward the gates, instead of stopping in the proper position, it would come against one of these chains instead of ramming the guard gate of the lock, so that a double protection is provided for the main gates. These chains, instead of being rigidly attached are connected with a system of hydraulic cylinders located in both side walls. These hydraulic cylinders, which are illustrated in Fig. 9, are so designed that they allow the chain to yield when it is subjected to a heavy load so that the forward movement of a ship will be checked gradually. The same mechanism is also used for raising or lowering the chains.

Referring to Fig. 9, there is a stationary outer cylinder A, into the lower end of which is inserted a movable combination plunger-cylinder B, which, in turn, fits over a stationary plunger C resting on the bottom of the pit. Two sheaves D and D_1 are connected with the plunger-cylinder by means of eye-bars E and E_1 and there are two similar sheaves at the top, the upper pair being at right angles to the lower ones. The chain enters the lock wall and is supported by an idler at G and, after passing around the sheaves referred to, is securely fastened to one of the beams at the top of the pit. In order to raise the chain, the plunger-cylinder B is moved downward by introducing water into the top of the outer cylinder, whereas the chain is lowered by forcing cylinder B upward. This upward movement is obtained by admitting water through the stationary plunger C. The maximum pressure required for raising or lowering the chain is about sixty pounds per square inch, and the water is supplied by a centrifugal pump. The resistance as the chain pays out, is obtained by means of a resistance valve designed to maintain a constant predetermined stress or pull on the chain. When the chain is struck by a ship the water pressure in the outer cylinder A rises, and when it has increased to 750 pounds to the square inch the resistance valve opens and allows enough water to escape to keep the pressure constant until the vessel is stopped. This pressure of 750 pounds corresponds to a stress of 100 gross tons on the chain. The valve for regulating the pressure is a differential piston type and its movement is regulated by an adjustable helical spring. When the pressure in cylinder A does not exceed 60 pounds the valve remains seated, but in case the chain is struck by a vessel, thus causing the pressure to rise rapidly, the valve opens as soon as the pressure has increased to 750 pounds. With this ingenious mechanism, a 10,000-ton ship having an initial speed of four knots an hour can be stopped within 73 feet, which is less than the distance between the chain and the gate. When the chains are lowered to permit the passage of a vessel they lie in cross grooves formed in the concrete bottom of the lock. Each of these chains weighs 24,000 pounds and the links are forged from 3-inch stock.

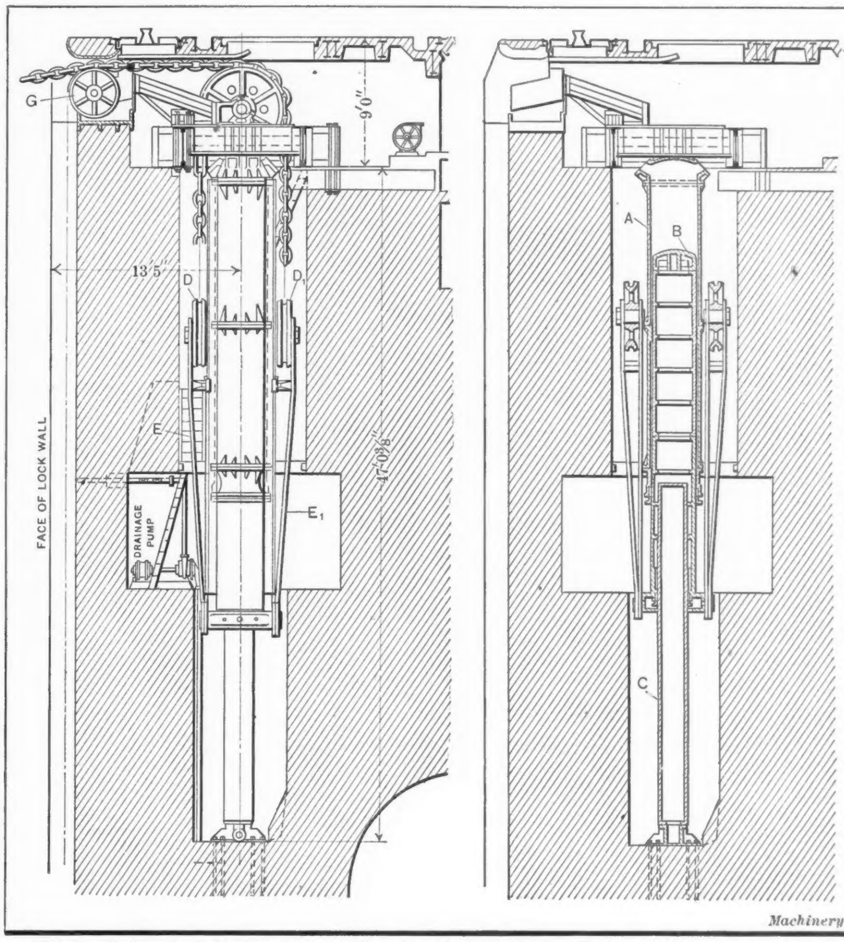


Fig. 9. Hydraulic Cylinders for operating the Massive Fender Chains which are to protect the Gates from being rammed by a Ship

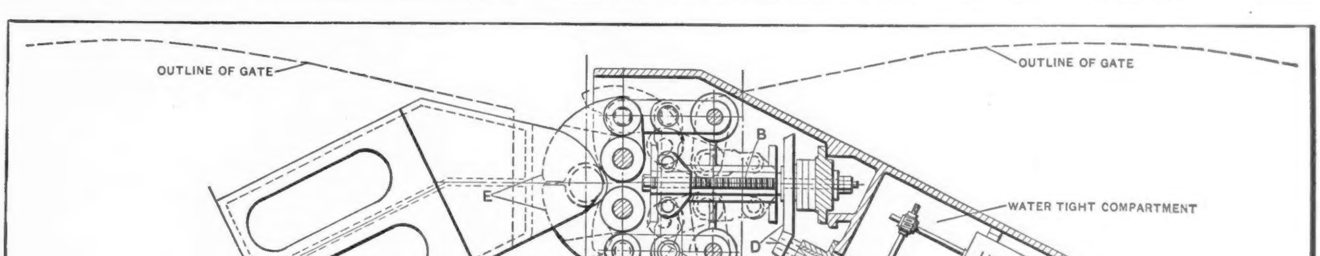


Fig. 8. Gate Locking and Miter Forcing Machine

Electric Towing Locomotives

As previously mentioned, electric locomotives will be used for towing all vessels into and through the locks in order to have a reliable and positive control of a ship's movement. In other canals, where ships enter the locks by their own power, a large percentage of the accidents have been due to the lack of control within the narrow limits of the lock chamber; hence towing locomotives will be used at Panama to obviate accidents of this kind. Ordinarily, four locomotives will be used, although the number will depend somewhat upon the size of the vessel. When four are used there will be two in front and two others behind, those ahead being used for towing and those in the rear for retarding the ship's movement when necessary. With this arrangement a vessel is under control laterally as well as longitudinally.

These locomotives are to operate on tracks laid on the central and side walls, there being two towing tracks for each flight of locks. In addition there is a return track on each side wall and one on the middle wall which is common to both towing tracks on this wall. All of these tracks run continuously the entire length of the lock and for some distance beyond on the approach walls at each end. The towing tracks have a central shrouded rack which is engaged by the driving gears of the locomotive, thus giving a positive and powerful movement. The speed when towing will be two miles an hour. After a ship has been drawn through the locks the locomotives will be switched over upon the return tracks, which are without racks, except upon the steep grades that lead from one lock level to another. The return speed is five miles an hour, except when ascending the steep grades just referred to.

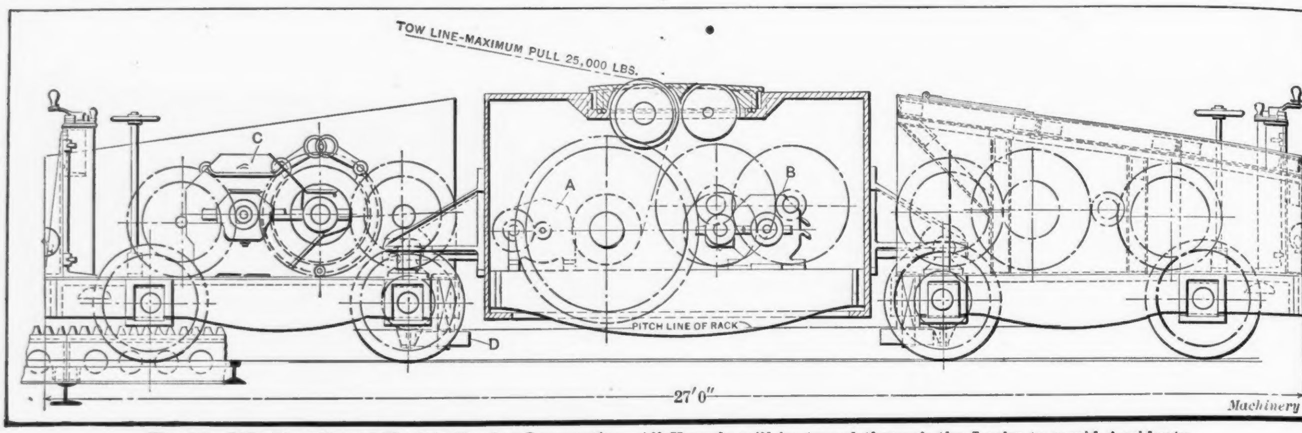


Fig. 10. Side Elevation of Electric Towing Locomotive—All Vessels will be towed through the Locks to avoid Accidents

An elevation view of a towing locomotive is shown in Fig. 10. It consists of three sections connected by universal joints. The front and rear parts are the tractors and the central element is equipped with a slip-drum towing windlass and hawser. Each tractor has a 75 H. P. driving motor with independent control from either end of the locomotive. The motor-driven towing windlass is so arranged that the hawser can be taken in or paid out and a pull exerted without any movement of the locomotive along the track. This feature is especially desirable for arresting a ship's movement or for changing the length of the tow line when, for example, the locomotive ascends to the level of the next lock while the water levels in the two chambers are being equalized and the ship necessarily remains stationary. The maximum pull on the tow line is fixed at 25,000 pounds, and whenever that amount is exceeded the drum slips, so that the locomotive and windlass are protected against excessive strains. The side pull of the towing line is taken by horizontal thrust wheels which bear against the side of the track. The electric current for operating these locomotives is conveyed through conduits at the side of the track. The electrical power for operating all the mechanism of the canal will be generated eventually at a hydro-electric plant located at the spillway of the Gatun Dam. There will also be a reserve station at Miraflores operated by steam. The two stations are to be connected by a transmission line, and current from either can be used.

The Emergency Dams

The emergency dams are one of the most interesting features of the locks. These dams are large bridge-like structures (see Fig. 11), two of which are installed in the forebay at the upper end of each set of locks, there being one dam for the flight of locks on the right side and another for the flight on the left. They are designed to close the upper ends of the locks, either to permit making repairs or in case free communication should be established between the two levels above and below a lock as the result of some unusual and unexpected accident. The emergency dams are arranged somewhat like a swing or center-pivot drawbridge, and, when in their normal position are parallel with the locks. In case it is necessary to use a dam the drawbridge is turned one-fourth a revolution so that it extends across the lock. Twelve equally spaced wicket girders *A* which are held in a horizontal position beneath the bridge, as shown in Fig. 12, are

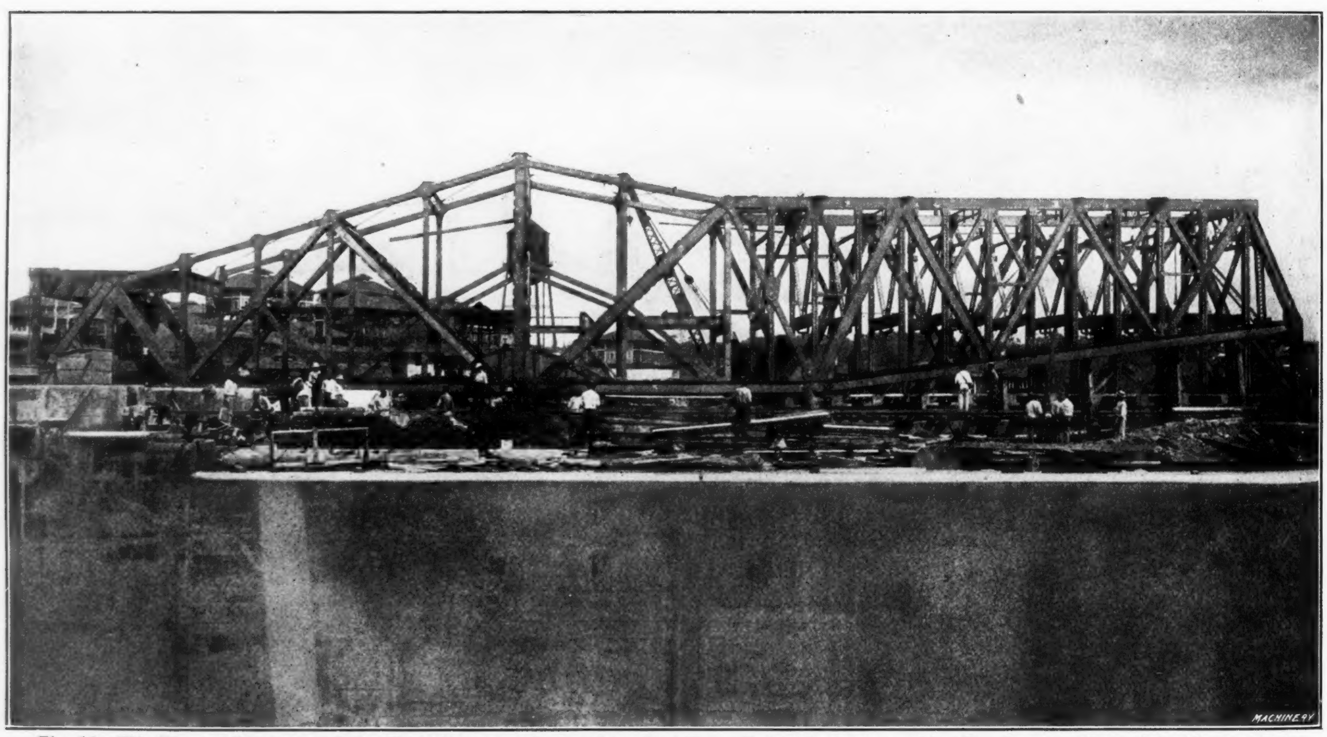


Fig. 11. Side View of Emergency Dam—These Dams are designed to close Upper End of Locks for making Repairs or in Case of Accident to Gates

then lowered as indicated in Fig. 13, both of these illustrations representing end views. These girders *A* are pivoted at *B* and the outer ends are lowered by wire cables *C*, which run through sheaves and are operated by motors upon the bridge. When the girders are down the lower ends are supported by cast buttresses fixed in the floor of the lock. A framework is now formed by the twelve girders which have been lowered for supporting the heavy flat roller gates *G* that are used for building up the emergency dam. There are thirty of these gates which, when in position, are arranged in five horizontal rows, each containing six gates. The dam is built up in sections and the girders form runways upon which the gates are lowered. At first the six bottom gates forming the lower row are let down, thus shutting off nine feet of water, and then the remaining four rows are lowered in succession. In this way the water, which would be rushing rapidly through the lock in case there were free communication between the upper and lower levels, would be gradually checked. In order to lower the gates readily they are mounted upon flanged wheels which run on heavy rails attached to the front faces of the girders.

After the dam has been formed there will be, of course, some seepage between the different sections, but as the rush of water has been checked, another dam can now be made

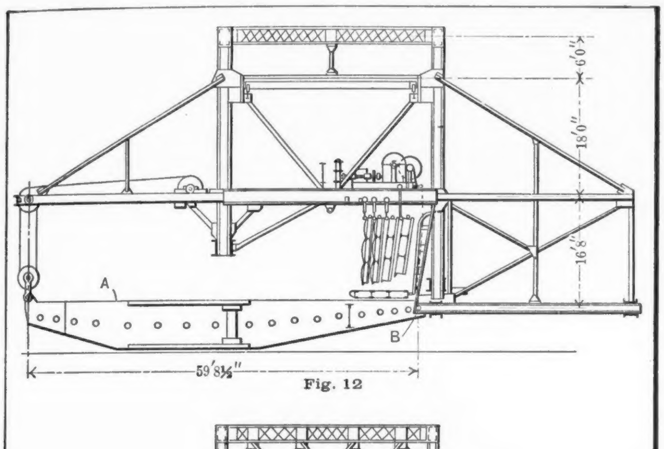


Fig. 12. End View of Emergency Dam, with Girders upon which Dam Gates are lowered, in Raised Position. Fig. 13. Girders and Gates lowered to form the Dam

by sinking a steel caisson above the temporary emergency dam. This floating caisson would be towed by the electric locomotives against an abutment at the upper end of the lock chamber. At this point the caisson, which is equipped with motor-driven pumps, is sunk by filling it with water, thus forming a secure dam. The emergency dam can then be raised, if necessary, for repairing the lock. When raising the gates *G*, preparatory to swinging the bridge back to its parallel position, four of the five rows are drawn up and suspended upon the bridge; then the girders with the lower row of gates still in place are raised to the position shown in Fig. 12.

The swing bridge of the emergency dam turns upon a central disk bearing which is interesting from a mechanical viewpoint because of the high pressure to which it is subjected. It consists of an upper casting attached to the bridge,

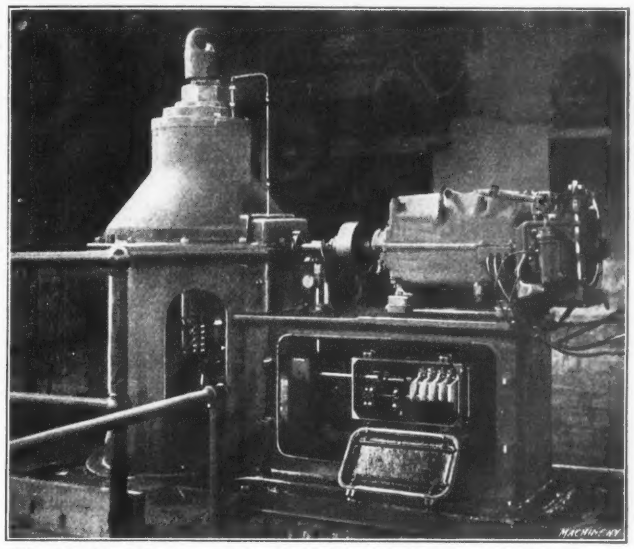


Fig. 14. Machine for operating Cylindrical Valve of Lateral Feed Culvert

three intermediate disks and a lower casting imbedded in the concrete pier. The disk which bears directly against the upper casting is flat on top and concave underneath, and the lower disk is similar but reversed in position. Between these concave disks which are made of forged chrome-vanadium steel there is a forged manganese-bronze center disk having convex surfaces which fit accurately into the concave disks above and below. These three disks are 43 inches in diameter and are subjected to a unit pressure of 4620 pounds per square inch, which is said to be the highest pressure that has ever been used in a similar location.

The bridge is turned by two motors each of which operates a separate pinion engaging a quadrant rack secured to the foundation. The driving end of the bridge contains a heavy concrete counterweight and is equipped with wheels running upon a circular track. There are also wheels and a circular track around the central bearing, but these wheels are about 1/16 inch above the track and are merely intended to steady the structure while it is being turned. Ordinarily the bridge is held in a fixed position by motor-operated wedges located at both ends and in the center. Of course, the emergency dams will rarely if ever be used, except during tests. The floating caisson gates previously referred to will also be used to close the tail-bay of a lock flight whenever the lower guard gates need to be cleaned, painted or repaired, and the caissons will probably be more satisfactory than the dams for closing the upper ends of the locks, except in cases of emergency.

Mechanism for Operating the Culvert Valves

The machinery for operating one of the gate valves which control the flow of water into the main feed culverts, is shown in Fig. 15, which is a side elevation. The top of the valve is seen at *A*. This valve is connected with the raising and lowering mechanism above by valve stem *B* which passes through a stuffing-box in a water-tight bulkhead *C*. The upper end of the valve stem is attached to a crosshead *D*, which is moved vertically by two non-rising screws *E*. These screws are rotated for opening or closing the valve, by a 50 H. P. motor *F*, which transmits its motion through reduction gears. Each screw is suspended from double roller bearings at the upper end and the crosshead is also guided by rollers running upon rails imbedded in the concrete. When the valve *A* is in motion the pressure of the water is transmitted

to trains of rollers *G* located on each side. As these rollers should move at half the speed of the gate, their retaining frames are positively raised and lowered by vertical stems *H*, which are connected by chains *J* with the crosshead *D*. The sheaves over which these chains pass are so arranged that rods *H* and the roller trains move at one-half the velocity of the gate valve. These valves can be operated locally, but ordinarily they will be controlled from a central station, as will be described more fully later. There is also auxiliary hand apparatus for closing the valve in case the machinery should fail when the valve is in the raised position.

One of the machines for operating a cylindrical valve, such as are used for the lateral culverts, is illustrated in Fig. 14.

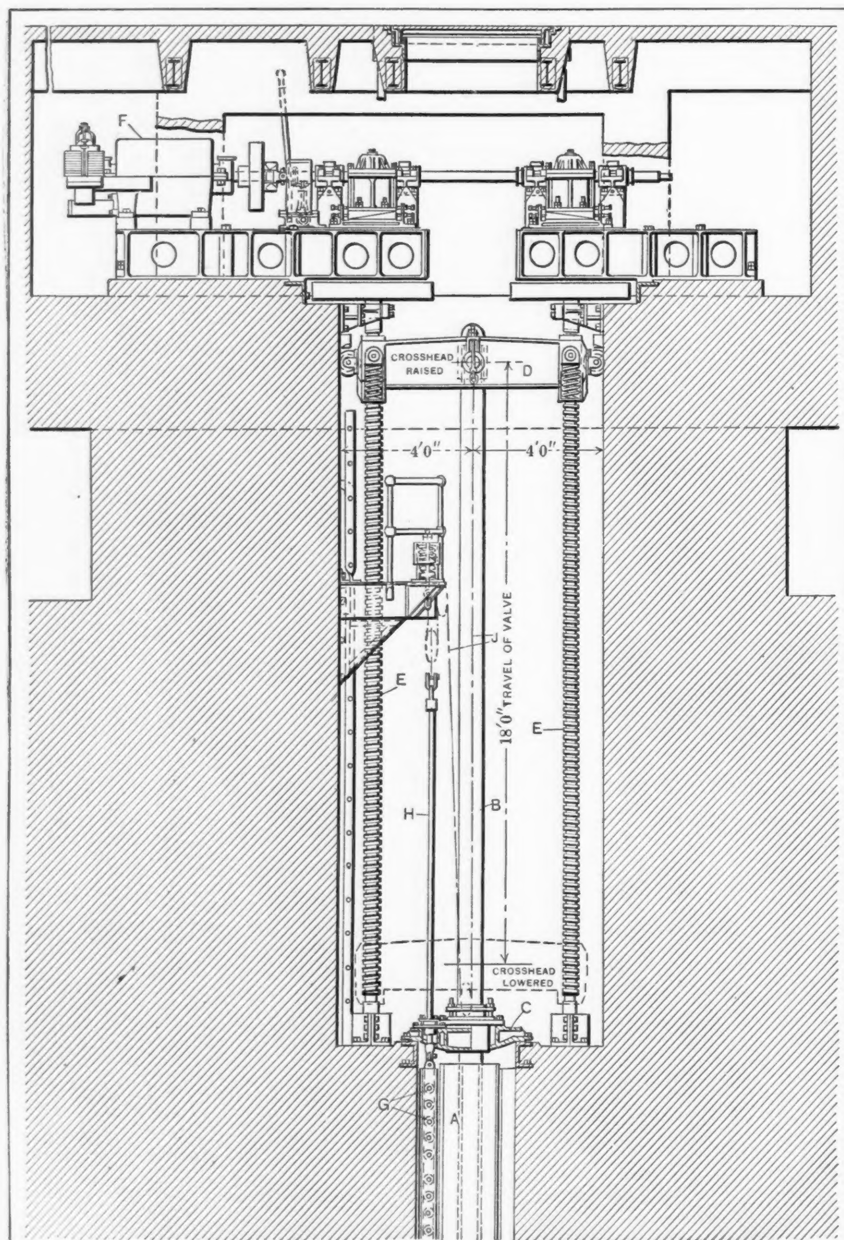


Fig. 15. Machinery for operating Stoney Gate Valves which control Flow of Water to Main Feed Culverts

There will be 120 of these machines in the six twin locks of the canal, there being one for each valve. These machines are located near the top of the center wall in chambers connecting with passage *A*, Fig. 3, whereas the valves *E* are down at the junction of the lateral and main culverts. The movable drum of the valve is connected by a stem rising vertically through a shaft in the masonry, with a tubular extension *A* (see Fig. 16), having a nut *B* which engages a non-rising, revolving screw *C*. This screw is turned through suitable gearing by motor *D*, and the travel of the valve in either direction is stopped at the proper point by a limit switch *E*. One of these valves can be opened or closed in ten seconds, and the large gate valves can be opened or closed in one minute.

Control of the Lock Mechanism

All mechanism that has to do with the actual operation of the locks, such as the culvert valves, gates and fender chains, will be controlled from a tower located on the highest lock in each flight. Owing to the number of gates, valves and chains to be operated, especially at Gatun, there will be many switches in the control tower, but all danger of operating the wrong switch will be avoided by an ingenious interlocking system which makes it impossible to throw the switches, except in the proper order or sequence. As a machinist might tersely explain it, the control will be "fool proof"—a very important feature in view of the fact that these locks are rather high in price and not easily repaired when once put out of commission. In order to pass a large ship through the locks at Gatun, it will be necessary to lower four fender chains, operate six pairs of gates and their locking mechanism, open and close eight pairs of main-culvert gate valves and thirty cylindrical valves. To accomplish all this, at least 98 motors will be set in motion twice during each lockage of a single ship, and this number may be increased to 143 under certain conditions.

The proper control of all these parts will be made easy by a unique "switchboard" located in the control tower. This switch- or control-board will be in the form of a bench or table 32 inches high, and will be in duplicate to conform to the twin locks. On this control-board there will be parts representing the gates, valves, etc., and indicators to show the height of the water in the big locks outside, so the towerman can see at a glance the exact position of the gates or other parts that are under his control. In order to further simplify the operation of the various parts, the switch which controls a certain gate or valve will be mounted on the board opposite that particular gate or valve. Hence, when a vessel is passing through the locks, the operator begins throwing switches at one end and moves along from one switch to another throughout the entire length of the board, the switches being operated successively and in the proper order. As each switch is thrown, the operation of that particular part of the lock is reproduced on the model, thus avoiding all confusion. The length of the board at Gatun is limited to 30 feet and the length of the other boards will be proportional to the length of their respective locks.

The interlocking system will be so perfected that all dangerous combinations in the relative positions of gates, valves, etc., will be impossible. To illustrate, the system will be so arranged that one of the gate valves in the main culvert could not be opened, unless the valves at the other ends of the locks, both upstream and down-stream, were first closed. This feature limits the operator to the act of equalizing the water levels in adjacent lock chambers and prevents him from allowing water to flow, say from the lake directly to the middle lock, thus eliminating the danger of flooding the lock walls and machinery rooms. The relative positions of the gate valves and gates will also be governed by the interlocks, which will prevent the valves from being opened a lock length above or below a gate that is being opened or closed. Hence, there can be no flow of water while the gates are in motion. The interlocking system for the cylindrical valves of the lateral culverts will be so designed that the valves on one side or the other will always be closed, except when cross filling from one twin chamber to the other, in which case both sets of valves can be opened at the same time by a special procedure. The interlocks will also make it necessary to unlock the miter forcing machine (illustrated

in Fig. 8) before the motors of the gate-moving machines can be operated, and they will even prevent such a mistake as leaving a fender chain down when it should be up guarding its gate, so that the locks are to be proof against carelessness and the machine will be the boss. These ingenious protective devices constitute one of the great features of the canal, although they may not be as spectacular and impressive as the cut at Culebra, the wonderful dam at Gatun or the gigantic locks.

* * *

A MACHINERY DEAL

BY A. P. PRESS

They were clearing out a lot of machines from the east wing, and replacing them with up-to-date automatic machinery. The foreman saw there would be a lot of tools to sell, and so he reasoned it out this way:

"They will go cheap as dirt—there are so many of them; and if I can get into the deal there will be some money in it." So he went over to a friend of his, and said:

"Charley, they are going to sell a lot of tools down to the A. B. Co., and while I don't dare to bid on them—in fact, it won't do for me to bid—still I would like to have a finger in the pie, and if you will go over and make a deal with the master mechanic, and buy them, I'll furnish the cash. Then we'll whack up on what we make. You can take them out to your place and store them, and sell them off. We ought to do a pretty good thing."

"All right, Charley, I'm yours."

The next day the foreman was bunching the tools off the way in a corner of the room. Skilton, the master mechanic, came in.

"That's a pretty good bunch of lathes we've got over there."

"Yes, it is, Mr. Skilton."

"What had we better do with them?"

"Well, I don't know, Mr. Skilton; seems to me it would be best to dispose of them. Since we got in the new automatics I can't see that we'll ever have any use for them."

"Well, I guess you're right. Say they are pretty good lathes to go as scrap. Tell you what I think. I would be glad to buy them, but of course it would hardly do. Why don't you buy them? I am willing to furnish the cash. It ought to be easy to dispose of them."

Poor Charley! He turned all colors of the rainbow.

"Well, sir, to tell the truth, I did think of that, and I spoke to a friend of mine, and he said he would be willing to buy them, and so we thought we would make a bid on them."

"Yes," said Mr. Skilton, "but I don't like to be left out in the cold on the deal. I am willing to put in half the cash and share one-third the profits."

A look of relief came over the foreman's face, as he found himself getting out of the hole, even though he lost one-half of his profits.

"All right, Mr. Skilton."

"Well you send your friend up to my office to-morrow, and we will finish up the sale."

Charley went up to the master mechanic's office the next morning, and as he told me afterwards, it was as good as a play.

"There I was buying Skilton's lathes from Skilto , and paying for them with Skilton's money, and while the purchasing agent had to be called in, he was only a side issue."

To cut the story short, Skilton sold the lathes, and also furnished the cash that he sold them for, and turned them over to Charley, and when I saw Charley last he was trying to unload them. So far he had not made much progress, and so we bought them of him.

We have done a lot of straightening out in our day, but we think this was the crookedest deal in the mechanical line that was legally straight, we ever had anything to do with.

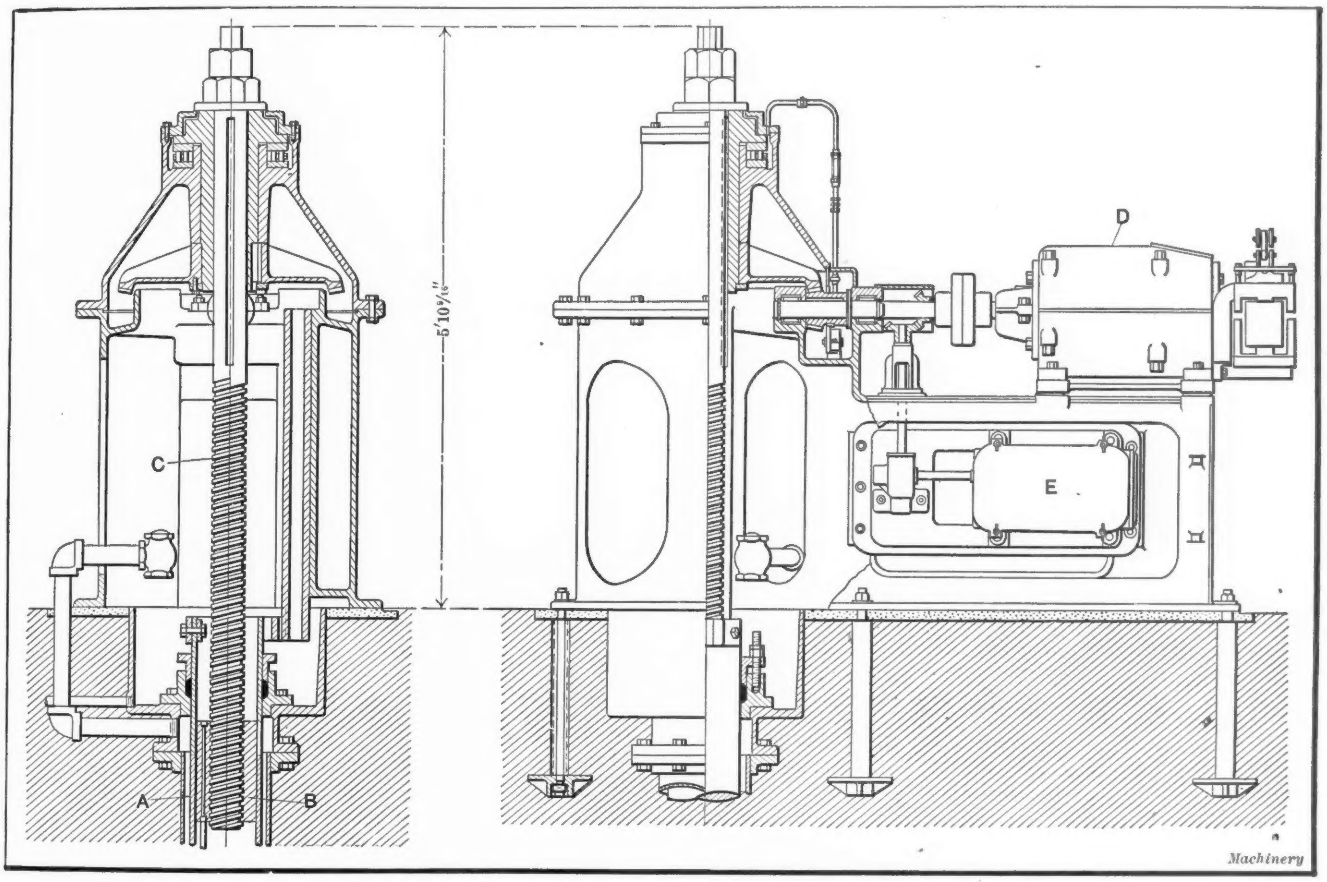


Fig. 16. Side Elevation of Cylindrical Valve Machine illustrated in Fig. 14

* * *

ACID-RESISTING ALLOY

The following alloy is claimed to possess exceptional qualities with regard to its ability to resist the action of acids: Nickel, 66.6 per cent; chromium, 18 per cent; copper, 8.5 per cent; tungsten, 3.3 per cent; aluminum, 2 per cent; manganese, 1 per cent; titanium, 0.2 per cent; boron, 0.2 per cent, and lithium, 0.2 per cent. This alloy is difficult to cast because it contracts considerably at the point of solidification. It can, however, be drawn into wire and is easy to work.

FREE IMPORT OF MATERIALS FOR SHIPBUILDING

In Section 5 of the Panama Canal Act a provision is contained as follows: "That all materials of foreign production which may be necessary for the construction or repair of vessels built in the United States, and all such materials necessary for the building or repair of their machinery, and all articles necessary for their outfit and equipment may be imported into the United States free of duty under such regulations as the Secretary of the Treasury may prescribe."

In giving effect to this provision, the United States Treasury regulations provide, in part, as follows:

Materials.—The word "materials" is defined as including any imported merchandise suitable for use in the construction or repair of a vessel or of its machinery to be incorporated therein after having undergone a process of manufacture subsequent to importation, or in its condition as imported, provided it has been purchased in the open market and was not constructed or fabricated upon a special order or after a special design. This will include raw materials, such as pig iron and lumber, to be worked up into the finished state in which they will enter into the hull or machinery, rough forgings and castings, but not finished ones, nuts, screws, bolts, steel plates, ship's knees, flooring, and other things which, though completed articles, are useful as parts in the construction of something else.

Articles.—The word "articles" is defined as including only such things as are suitable for use in their condition as imported in the outfit and equipment of a vessel. However, articles will be considered as suitable for use in the condition in which imported, although they are to be fitted, polished, painted, or otherwise improved in condition or fixed in place subsequently.

Outfit and Equipment.—The term "outfit and equipment" is defined as including portable articles necessary or appropriate for the navigation, operation, or maintenance of a vessel, and not permanently incorporated in its hull or machinery, and not constituting consumable supplies. The term includes, therefore, anchors, chains, cables, tackle, boats, repair parts, life-saving apparatus, wireless telegraph apparatus (except the motor generator), nautical instruments, searchlights, signal lights, lamps, furniture, carpets, table linen, tableware, bedding, arms and munitions, and also articles to be used in renewal or replacement of articles of original outfit and equipment.

Machinery, including all the propelling mechanism, and also auxiliary machinery permanently incorporated in the hull of the vessel, such as pumps, steering gear, lighting plants, refrigerating plants, steam winches, hoisting engines, generators, motors, condensers, feed-water heaters, evaporators, switchboards, etc.; provisions, wines, coal, medicines, and other similar consumable articles, and materials intended to be manufactured into articles of outfit or equipment, will not be admitted free of duty under this Act. Neither materials for the construction or repair of vessels or of their machinery, nor articles for outfit and equipment, will be admitted free of duty when imported to be kept in stock for sale.

Materials for construction or repair of vessels, or of their machinery, and articles for outfit and equipment, may be entered for warehouse, and if withdrawn within three years from the date of importation for use in such construction, repair, outfitting, or equipping, will be entitled to enter free of duty upon such withdrawal.

The *Practical Engineer* (London) comments as follows:

"The 'definitions' of the act leave very little of the act intact. The key to the matter is, however, simple:—(1) Promise free admission on broad general lines subject to 'regulations.' (2) Draft the regulations to include and exclude anything you please. Classifications are no difficulty. If you want to include searchlights and exclude evaporators call the searchlight an 'article' and the evaporator a 'machine,' or *vice versa;* it matters not. Lighting plant can be cut in pieces. If you fear competition in some parts, call the motor a machine to keep it out. The rest of the plant can be called an article to let it in, but, as this will admit switchboards, in which competition cannot be entertained, double back and call the switchboard a machine."

STRENGTH OF WIRE ROPE DRUMS

BY R. H. CREVOISIE*

In the following are given formulas which will be found useful when calculating the strength of wire rope drums. Usually if a drum is designed to be strong enough to resist crushing, it will be strong enough to resist bending also, except in cases of extremely long drums. To be on the safe side, however, the strength should be calculated for both crushing and bending. Let us first consider the calculations to find the crushing stresses. Let

T = tension in one rope, in pounds;
t = thickness of drum in inches at bottom of groove;
P = pitch of scoring, or distance between grooves in inches;
L = span in inches from center to center of bearings;
D = diameter at the bottom of the groove;
d = inside diameter of drum;
W = total load in pounds;
M_b = bending moment.

We can then calculate the total crushing stresses as follows:

$$\text{Section modulus} = Z = 0.0982\left(D^3 - \frac{d^4}{D}\right);\quad M_b = \frac{WL}{4}$$

$$B = \frac{M_b}{Z};\quad C = \frac{T}{Pt}.$$

Combined stress $= \sqrt{B^2 + C^2}$

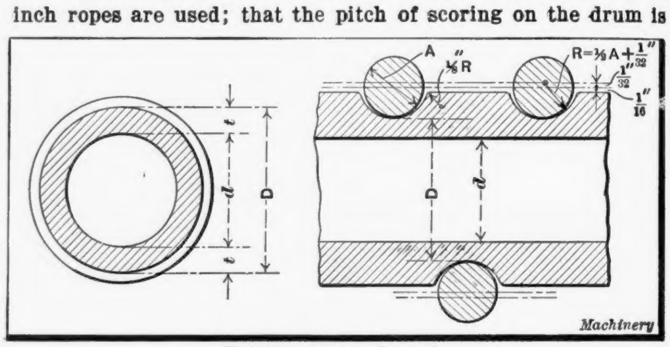
The combined stresses should not exceed 6000 pounds per square inch for cast iron.

To find the strength for bending, let S = safe fiber stress, which should not exceed 3000 pounds per square inch for cast iron.

The diameter of the hole through the drum may then be found from the following formula:

$$d = \sqrt[4]{D^4 - \frac{D \times M_b}{0.0982\,S}} \qquad (2)$$

As an example, assume that on a five-ton crane, two 7/16-inch ropes are used; that the pitch of scoring on the drum is



Notation used in Formulas

¾ inch; and that the distance from center to center of the bearings on the drum is 6 feet 2 inches. The inside diameter of the drum is 12 inches, and the diameter at the bottom of the grooves, 14 11/16 inches. From the formulas given we find the combined crushing stress:

$$Z = 0.0982\left(14.69^3 - \frac{12^4}{14.69}\right) = 172.6$$

$$M_b = \frac{10{,}000 \times 74}{4} = 185{,}000$$

$$B = \frac{185{,}000}{172.6} = 1072;\quad C = \frac{5000}{\frac{3}{4} \times 1.34} = 5000$$

Combined stress $= \sqrt{1072^2 + 5000^2} = 5113$ pounds per square inch.

As this is less than the allowable stress of 6000 pounds per square inch, the drum is safe for crushing.

To determine if the drum will withstand bending, use Formula (2), assuming the diameter at the bottom of the grooves to be 14 11/16 inches and solving for the inside diameter:

$$d = \sqrt[4]{14.69^4 - \frac{14.69 \times 185{,}000}{0.0982 \times 3000}} = 13.9 \text{ inches.}$$

Since the inside diameter is only 12 inches instead of 13.8 inches, it follows that the drum is strong enough for the purpose for which it is designed.

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MACHINE FORGING—1

MACHINES AND METHODS EMPLOYED IN THE MANUFACTURE OF BOLTS, RIVETS, NUTS, MACHINE PARTS, ETC.

BY DOUGLAS T. HAMILTON*

The bolt and nut industry in America started in a very small way in Marion, Conn., in 1818. In that year Mr. Micah Rugg, a country blacksmith, made bolts by the forging process. The first machine used for this purpose was a device known as a heading block, which was operated by a foot treadle and a connecting lever. The connecting lever held the blank while it was being driven down into the impression in the heading block by a hammer. The square iron from which the bolt was made was first rounded so that it could be admitted into the block. At first Mr. Rugg only made bolts to order, and charged at the rate of sixteen cents a piece. This industry developed very slowly until 1839, when Mr. Rugg went into partnership with Martin Barnes, and together they built the first exclusive bolt and nut factory in Marion, Conn. The bolt and nut industry started in England in 1838, the credit for which is given to Mr. Thomas Oliver, of Darlston, Staffordshire. His machine was built on a little different plan from that of Mr. Rugg's, but no doubt was a further development of the first machine. Mr. Oliver's machine was known as the "English Oliver."

As is generally the case with a new industry the methods and machines used were very carefully guarded from the public, and this characteristic seems to have followed this industry down to the present time, judging by the scarcity of available information on the subject. Some idea of the methods which were at first employed to retain all information in the factory in which it was originated, is well brought out by the following instance: In 1842 when the industry was beginning to be generally known, it is stated that a Mr. Clark, who at that time owned a bolt and nut factory in New England, and had devised a special machine for their manufacture, had his forging machine located in a room separated from the furnaces by a thick wall. A hole was cut through this wall, and the man who operated the machine received the heated bars from the furnace through the small hole in the wall. The only person who ever got a glimpse of the machine was the operator. The forge man was not permitted to enter the room.

Machine forging, as we know it to-day, is of wide application, embracing a large number of machines and processes that apply, in a measure, to almost any manufacturing plant. Machine parts hitherto made from castings are now made much more economically by the use of the drop-hammer or forging machine, and give much more satisfactory service. In the following articles the various forging machines and methods employed in the manufacture of bolts, rivets, nuts, machine parts, etc., will be taken up.

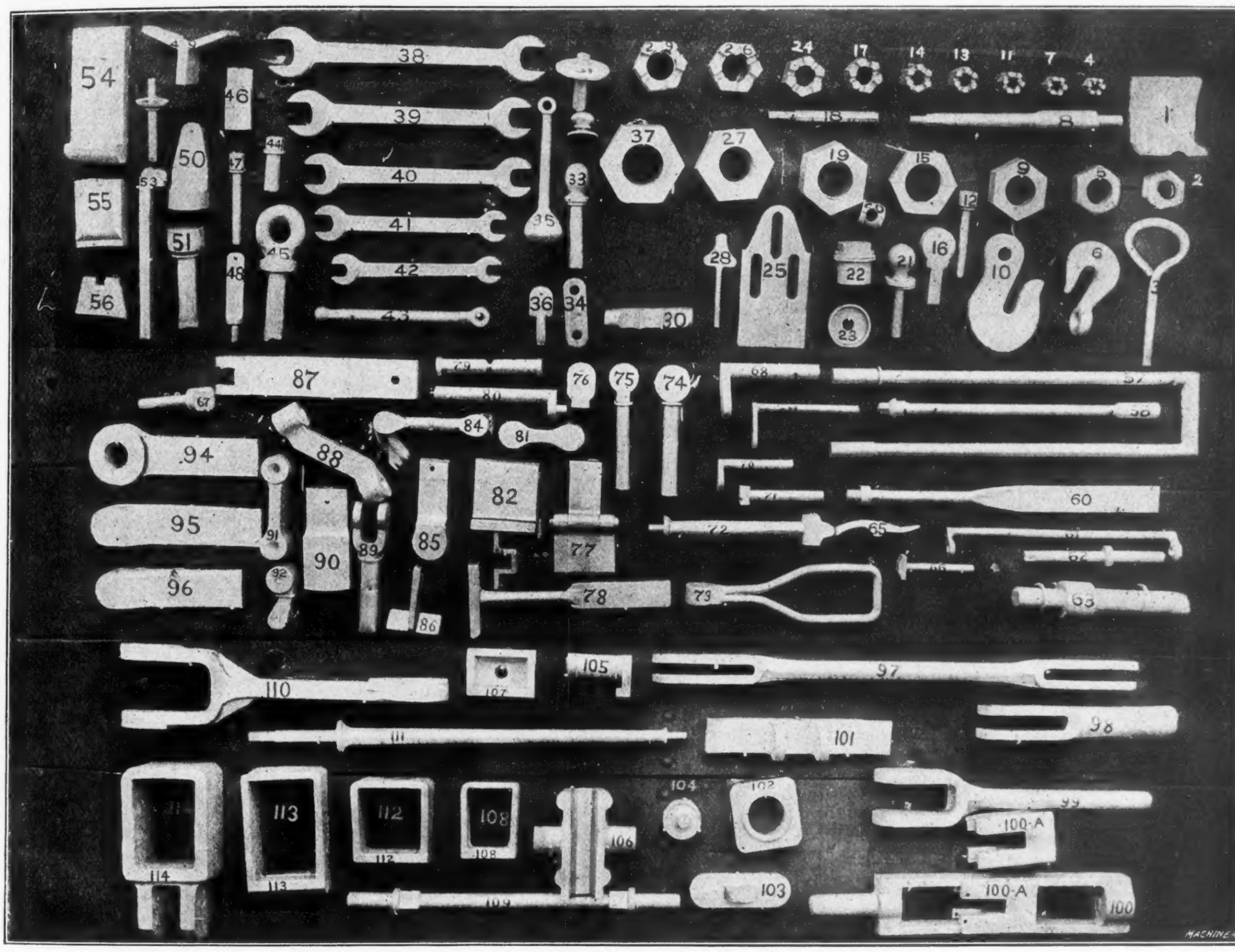


Fig. 1. Examples of Forged Parts produced in the Collinwood Shops of the L. S. & M. S. Railway on Ajax Forging Machines

Types of Machines

Upsetting and heading machines are divided into two general classes, namely, stop-motion and continuous-motion headers. The stop-motion headers have the greatest range, and are primarily used for heading bolts and all sorts of upset forgings. The continuous-motion headers are used only for heading rivets, carriage bolts and short lengths of hexagon and square head machine bolts; they produce these parts at a much faster rate than is possible with a stop-motion header, but their range of work is limited. The universal practice is to shear the bars cold when working a stop-motion header, and only in special cases, where the shank of the headed piece is very short, is the side shear used.

Rivets, etc., made by the continuous-motion header are made by the process known as "off the bar," that is, a bar is

* Associate Editor of MACHINERY.

heated for a distance of approximately four feet, and is then pushed into the machine where the moving die acts as a shear and cuts off the blank. The latter is immediately gripped against the stationary die, whereupon it is headed and ejected. This whole cycle of movements is accomplished in one revolution of the flywheel.

Operation of Plain Bolt and Rivet Machines

Briefly stated, a plain bolt and rivet machine comprises two gripping dies, one movable and the other stationary, and a ram which carries the heading tool. The heated bar is placed in the impression in the stationary gripping die, and against the gage stop; the machine is then operated by pressing down the foot treadle shown immediately in front of the machine in Fig. 2. As already mentioned the stock is generally cut to the desired length before heading, on this type of machine, especially when it is long enough to be conveniently gripped with the tongs; but it can be headed first and afterward cut off to the desired length in the side shear. It is also possible, in some makes of machines, to insert a cutting tool to cut off the blank before heading, when the work is not greater in length than the capacity of the machine.

There are several methods used in making bolts and rivets in a regular forging machine. In Fig. 3 is shown a diagrammatic view of a set of forging dies which have an extremely wide range of application. In this type of dies the head on the bolt is formed by rotating the bar between the gripping dies after each blow of the plunger. For a square-headed bolt, the bar is turned twice through a space of 90 degrees, and is generally given two or more blows in each position. A hexagon-head bolt usually requires at least six blows to complete one bolt, and the shape of the head depends to a large extent on the skill of the operator. The wide range of work, however, which can be handled in dies of this type, make them of almost universal application, especially in a railroad shop.

Fig. 4 shows a set of single-blow rivet dies which are used in a continuous-motion rivet header, and illustrates how these dies are operated in the production of a rivet in one blow. The heated stock is fed in and cut off to the exact length by a shear *A*; it is gripped between dies *B* and *C* while being cut off. The tool *D* held in the ram of the machine then advances, upsetting the head to the shape shown, whereupon the movable die backs out, allowing the formed rivet to drop out and the bar to be inserted to the stop, ready for the next piece. The type of bolt heading tool illustrated in Fig. 5 is known as a double-deck three-blow bolt die; its use and operation will be explained later.

Successive Steps in Heading Bolts

Figs. 6, 7 and 8 show the successive steps followed in the forging of a hexagon-head bolt in the type of bolt forging

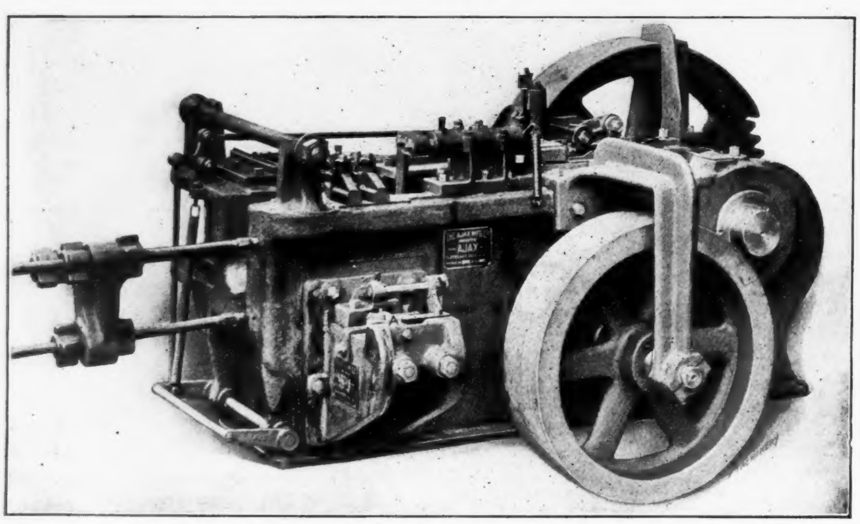


Fig. 2. Ajax Bolt Heading Machine suitable for all Types of Machine Bolts and Upset Forgings

dies illustrated in Fig. 3. Bar *A* which is heated for a portion of its length, is placed in the impression in the stationary gripping die *B*, as shown in Fig. 6, and is gaged to length by the lifting stop *C*. The machine is then operated, and the movable die *D* closes in on the bar, gripping it rigidly. The stop now rises and as the ram of the machine advances, the plunger *E* upsets the end of the bolt, the blocks *F* and *G* forming a flat on each side of the upset end. The operator keeps his foot on the treadle, and as the movable die backs out, he rotates the rod one-third turn. This operation is repeated until the head has been correctly formed. The operator now

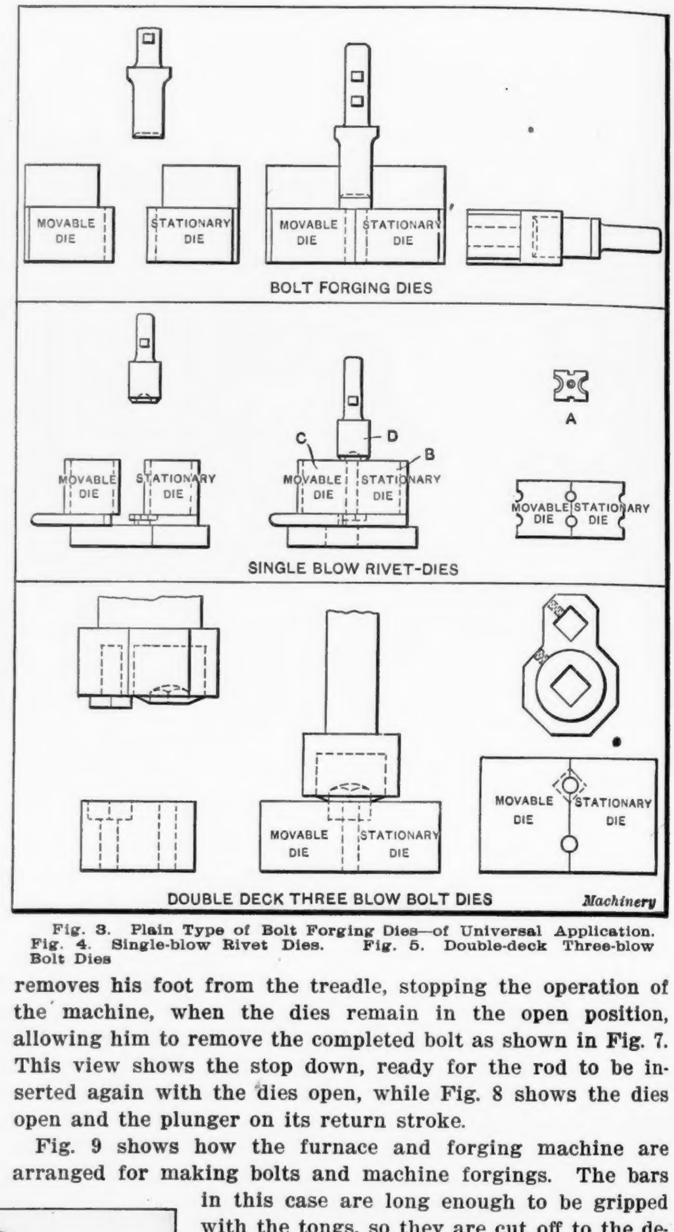


Fig. 3. Plain Type of Bolt Forging Dies—of Universal Application. Fig. 4. Single-blow Rivet Dies. Fig. 5. Double-deck Three-blow Bolt Dies

removes his foot from the treadle, stopping the operation of the machine, when the dies remain in the open position, allowing him to remove the completed bolt as shown in Fig. 7. This view shows the stop down, ready for the rod to be inserted again with the dies open, while Fig. 8 shows the dies open and the plunger on its return stroke.

Fig. 9 shows how the furnace and forging machine are arranged for making bolts and machine forgings. The bars in this case are long enough to be gripped with the tongs, so they are cut off to the desired length in a power shear before heading. From the power shear the bars are brought to the heating furnace in the truck shown to the right in the illustration, where one end of the bars is heated to the desired temperature. This furnace is heated by oil and is placed as close to the forging machine as possible. The man who attends to the heating of the stock places the rods in a row, and as soon as the end to be headed reaches the proper temperature, he quickly removes the heated bar and passes it to the forging machine operator, who immediately places it between the dies, operates the machine, and forms the head. In this particular example the bolt is 1¼ inch in diameter by 12 inches long, and is formed in three blows in double-deck dies of the type illustrated in Fig 11. The dies and heading tool are kept cold by means of a constant stream of water. As soon as the bolt is headed it is thrown in the truck to the left, which is used for conveying the bolts to the threading machines.

Fig. 10 shows a view looking down into the die space of

the "Ajax" bolt header, from which an idea of the relation between the working members can be obtained. The back stop *A* is used for locating the bar in the correct position. This stop is sometimes used instead of the swinging stop *B* in

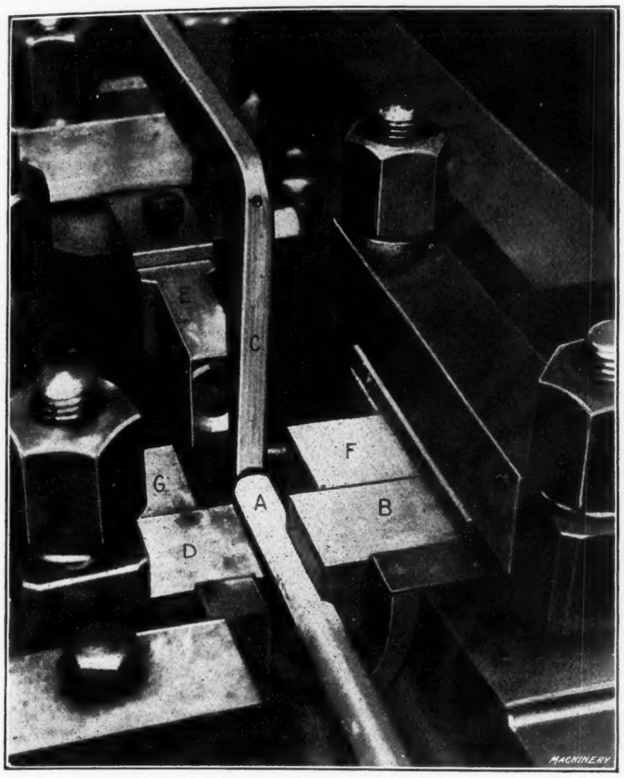


Fig. 6. First Step in the Production of Bolt Heads in Plain Forging Dies of the Type shown in Fig. 3

locating the work. This view also shows how the gripping dies are held in the die space; a heel plate fastened to the frame of the machine and to the movable die slide by studs

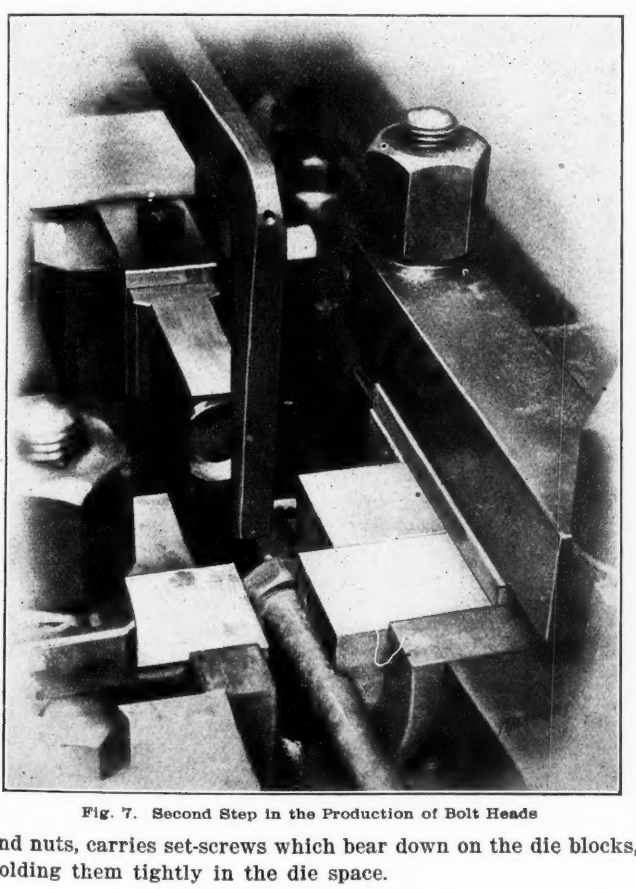


Fig. 7. Second Step in the Production of Bolt Heads

and nuts, carries set-screws which bear down on the die blocks, holding them tightly in the die space.

Types of Bolt Header Dies

Fig. 11 shows a type of bolt heading dies known as double-deck three-blow bolt dies, which are used for finishing hexagon-head bolts. The two gripping dies *A* and *B*, as a rule, are made from blocks of tire steel and each gripping die is made from three pieces to facilitate machining. The lower header punch *C* is cupped out to form a hexagon and is held in the heading tool-holder which is attached to the ram of the machine. The upper punch *D* is held in the same manner as the lower heading punch, and forces the bolt into the hexagon impressions in the dies after it has been roughly formed in the lower impressions. This type of die produces a bolt free from fins and burrs, and accurate as regards size and shape. The bolt is given one blow in the lower position and then raised to the upper die impression, where it is generally given two blows.

A combination double-deck gripping die for making square- and hexagon-head bolts is shown in Fig. 12. The construction of these dies is similar to those shown in Fig. 11, with the

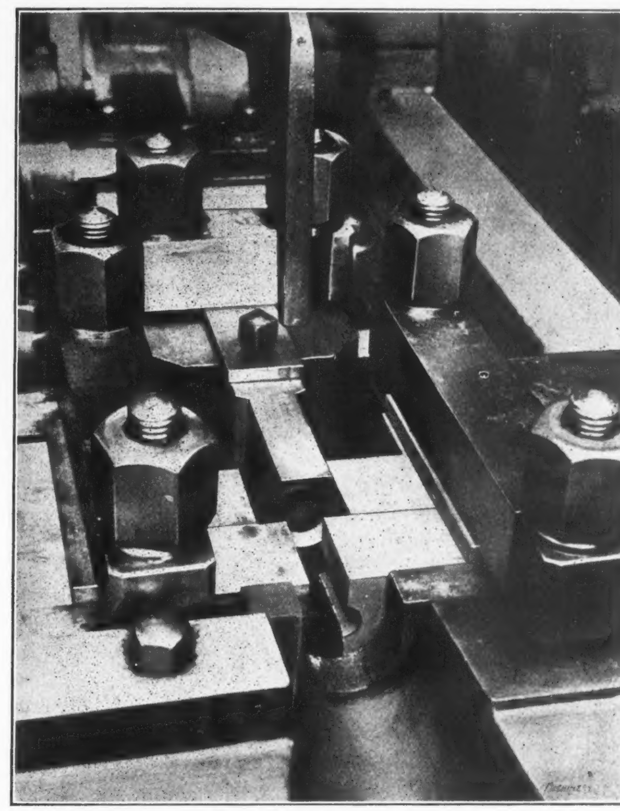


Fig. 8. Third Step in the Production of Bolt Heads

exception that these dies can be used for making both square- and hexagon-head bolts. The punches for forming the hexagon- and square-head bolts are shown at the right and left,

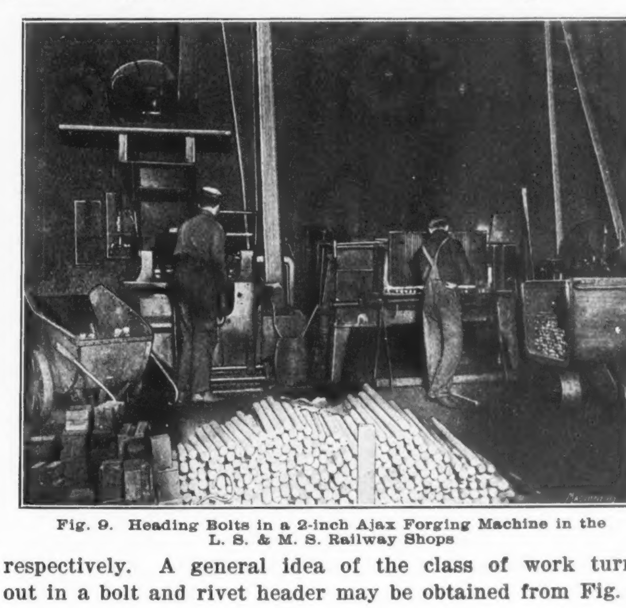


Fig. 9. Heading Bolts in a 2-inch Ajax Forging Machine in the L. S. & M. S. Railway Shops

respectively. A general idea of the class of work turned out in a bolt and rivet header may be obtained from Fig. 13.

Construction of National Wedge-grip Bolt and Rivet Header

Fig. 14 shows a view of a two-inch National wedge-grip bolt and rivet header which is used for making bolts, rivets and miscellaneous forgings. There are a number of inter-

esting features connected with this machine, one of which is the wedge-grip and automatic relief mechanism. In operating a bolt and rivet header it is necessary that the work be placed directly in the impression in the gripping dies and not between their opposing faces. Both these dies must come tightly

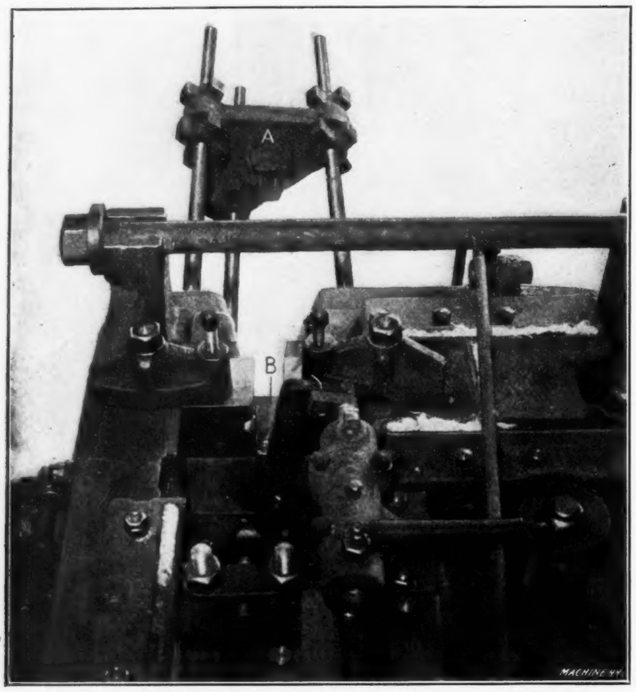


Fig. 10. View looking down into the Die Space of an Ajax Bolt Forging Machine

together and are made to do so by the mechanism of the machine, so that any foreign body preventing the correct movement of these dies is bound to cause trouble by breaking the machine.

Various methods have been provided for obviating this

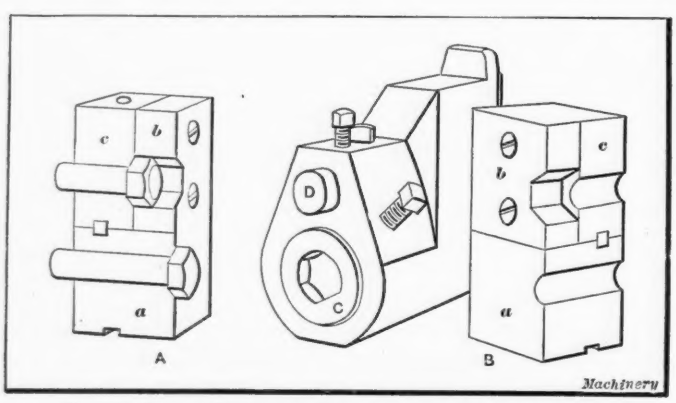


Fig. 11. Type of Double-deck Dies used in producing Bolt Heads without Fins or Burrs in Three Blows

difficulty, one of which is the interception of a shearing pin in the movable gripping die slide, which, when the foreign body is placed between the dies, is sheared off without causing any damage to the machine. Another method, which is a special feature of the National wedge-grip header, is a spring

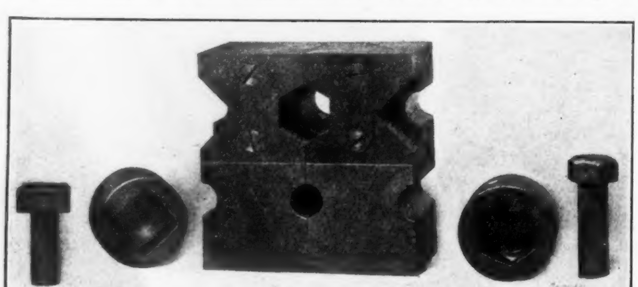


Fig. 12. Combination Square and Hexagon Double-deck Bolt Dies

relief, which throws the entire gripping mechanism out of action should the stock or any foreign body be caught accidentally between the dies and prevent them closing. The action of this relief is indicated in the diagrams Figs. 15 to 17. In Fig. 15 the gripping dies are shown closed and the relief mechanism does not operate. In Fig. 16, the gripping dies are shown open and the ram is at its extreme backward stroke; while in Fig. 17 the dies are open, but, as can be seen, the ram is at the forward end of the stroke. The latter view shows what happens when a foreign body is caught between the gripping dies and prevents them from closing.

The relief mechanism consists of a spring plunger *A*, the front end of which is beveled, and which is kept out by a coiled spring. This plunger, as indicated in Fig. 16, presses against the angular projection on the movable gripping slide. Now when a foreign body comes between the gripping dies and prevents them from closing, this spring plunger is forced back and the toggle joint operating the wedge-gripping slide remains stationary, which allows the dies to remain open, ut the ram to complete its full forward travel. This relief

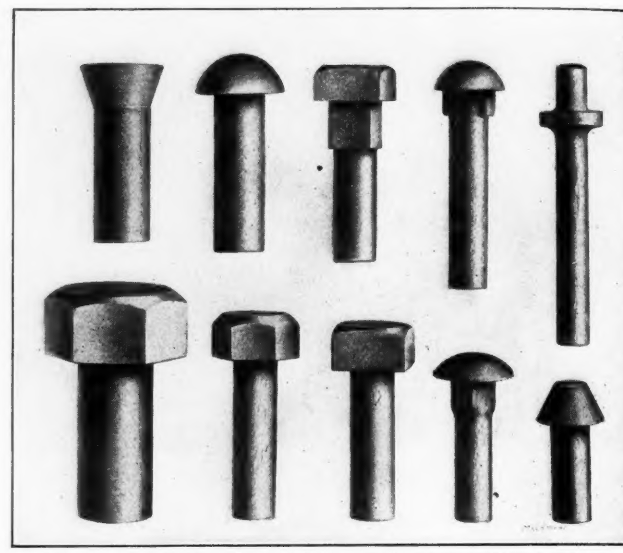


Fig. 13. Some Examples of Work turned out on National Wedge-grip Bolt and Rivet Header

will operate up to the time the dies are closed, but when the dies are closed, the gripping pressure is positive.

An important feature of this machine is the wedge-grip for the movable slide. This consists of a slide *B* to which the toggle lever is attached, and which is moved back and forth by the latter through the movement of the crankshaft. The forward end of the slide *B* is beveled and forms a solid metal

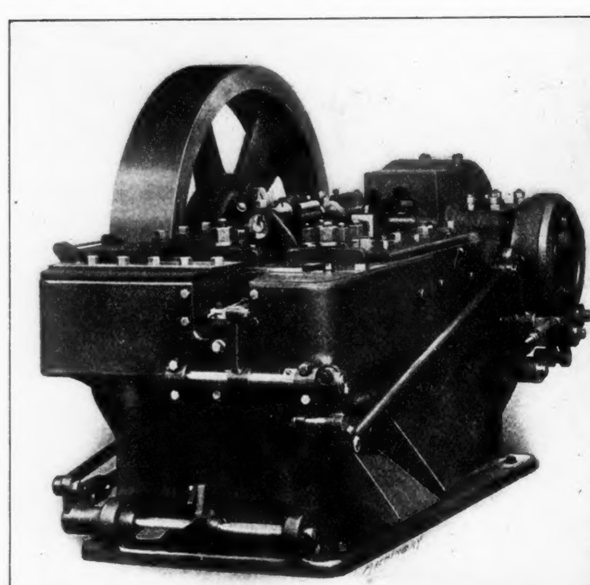


Fig. 14. Two-inch National Wedge-grip Bolt and Rivet Header

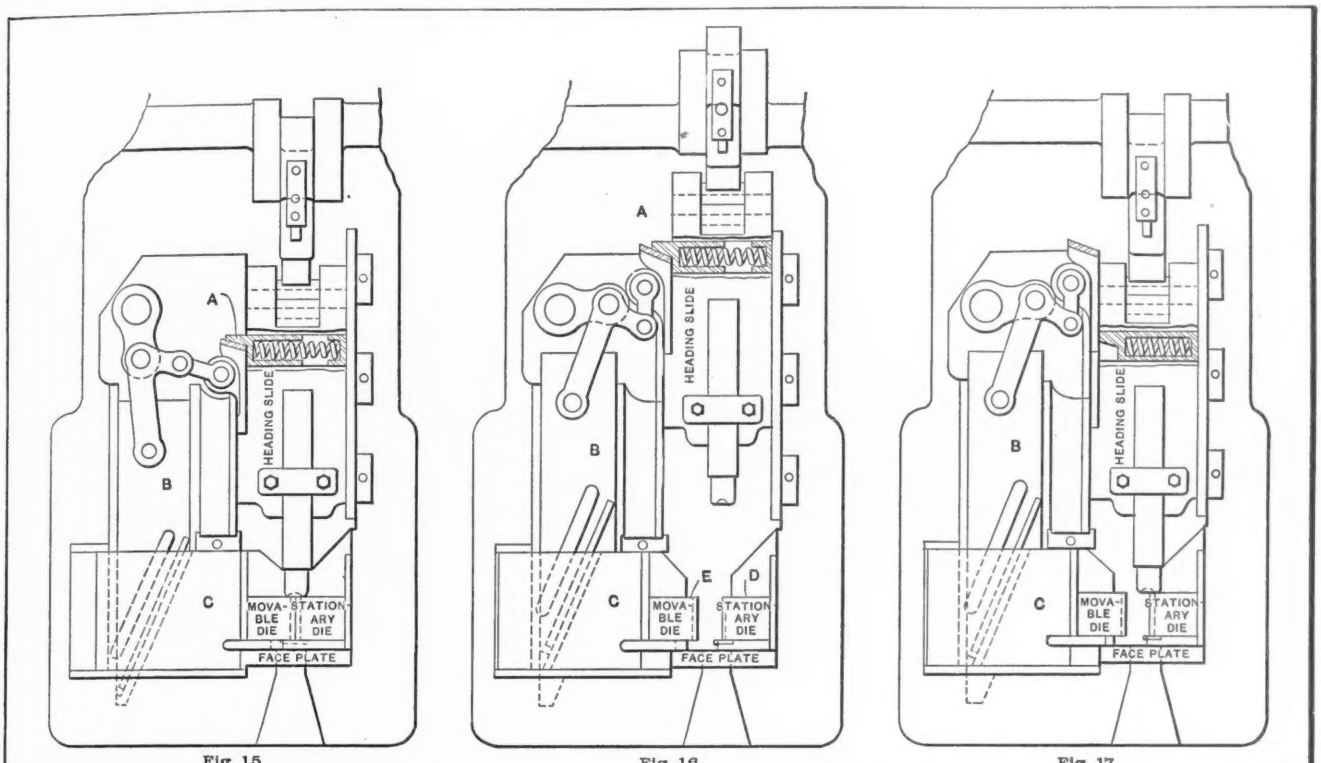
backing when the gripping slide *C* is in the forward or gripping position—when the dies are closed. This means of locking the movable die during the heading operation prevents any rocking or wobbling of the slide and causes an even pressure to be exerted over the entire working surface of the dies. The stationary die *D* and movable die *E* are set so that their working faces merely touch, and the rigidity of the

grip prevents any spring, so that the work can be produced without fins and burrs. By not having to set the dies ahead, the pounding or battering and premature wearing out of the dies is prevented.

Fig. 18 shows more clearly how the movable and stationary dies are retained in the die space, and how they are backed up by steel liners. This illustration also shows the shape of the sliding wedge and the steel liners that intercept it and the gripping slide. From an inspection of this illustration it will be seen that with this sliding wedge mechanism it is practically impossible for the dies to give or spring when in operation on the work.

* * *

Automatic screw machines which have been in constant use for years become worn in the slides and require readjustment. But readjustment sometimes may be delayed long after a theoretical consideration would require it to be made. The reason is that some cutting oils used deposit a gum which works in between the slides and partially fills the spaces made by wear. Hence it is inadvisable to thoroughly clean an automatic screw machine in all its working parts unless the machine is overhauled and wear is taken up by first scraping and then adjusting. A thorough cleaning removes the accumulated gum and leaves the parts too uneven to be adjusted closely to prevent shake.



Figs. 15, 16 and 17. Diagrams illustrating Construction and Operation of National Wedge-grip Bolt and Rivet Headers

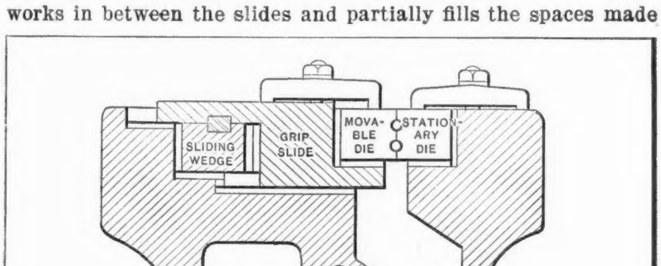
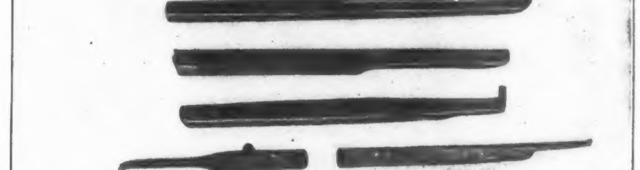


Fig. 18. Section through Die Box of the National Wedge-grip Bolt and Rivet Header

"SORE FINGER" BORING TOOLS

All machinists, of course, are familiar with the type of forged lathe boring tools shown in tthe accompanying illustration, but we wonder how many have heard them called "sore finger" boring tools? The credit for this expressive appellation is given to the late Edwin Reynolds, formerly chief engineer of the Allis-Chalmers Co., Milwaukee, Wis., by Mr. C. M. Conradson of Madison, Wis., who furnished the photograph.

A compilation of the expressive terms used by machinists to designate various machines and tools used in the manufacture and repair of machinery, would be interesting, no doubt, and we shall be glad to have readers from America and abroad send in odd terms used in the shops where they have worked to designate various equipment, especially when they are as pat as the foregoing example.



"Sore Finger" Boring Tools

* * *

With some people the fear of hard times seems to be a chronic disease, but during the last few months we have met some specimens of a new type of "worrier." There are some men who seem to actually fear that we are *not* going to have hard times during the next four years. We are speculating as to whether this is not merely because they are afraid that they will not have the satisfaction of saying, "I told you so."

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APRIL, 1913

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THE MODERN CONCEPTION OF SALESMANSHIP

Someone said long ago, "Anybody can sell anything to a man if he wants it. The real salesman is the man who can sell something to a man that he does not want." Fortunately this conception of the best class of salesmanship is no longer accepted by most high-class business concerns. Experience has taught that little is to be gained by selling something to a man that he does not want, and at the present time the best salesmen are not those who work on that theory, but those who have the ability to understand and analyze the customer's requirements and make him see his own needs, so that the sale means a bargain both to the buyer and seller. The most important thing for a successful growing concern is not merely to make sales, but to create a market, and this can be accomplished by selling to customers who actually want to buy—otherwise there is no likelihood of a second order from the same source. There is a difference between influencing a man to buy something he really needs and merely selling something to him. The salesman who makes men *buy* is a good salesman; the one who merely *sells* may in the long run lose his market.

* * *

SECRET HARDENING AND TEMPERING PROCESSES

That the hardening and tempering of steel tools is still largely a rule-of-thumb process is indicated by the fact that the circulars and other advertising matter of tool manufacturers abound in references to steel being tempered by secret processes. Shear blades are tempered in "our secret solution;" razor blades are hardened by "our secret method;" springs are tempered in "a bath prepared by a secret formula;" an inventor has "a secret process for hardening and tempering milling cutters without scale," and so on. One spring maker pays $5 a pound for a "secret salt" which he uses for a tempering bath. An expert metallurgist ventured the opinion that the chemical constituents were worth about sixteen cents a pound!

In view of the scientific work that has been done to determine exact methods of treatment for various kinds of steel, and the success with which scientific methods are attended, the claims made for secret processes are unworthy of this advanced age. It is of course true that all the vagaries of that complex compound known generally as "steel" have not been mastered by scientific experts, and certain so-called secret methods may be superior to more common methods within certain narrow limits, but that fact does not warrant the users in making the absurd claims for superiority that some do.

* * *

SAFETY IN THE MANUFACTURING INDUSTRIES

One of the gratifying features of the presidential address recently made by Mr. Arthur Williams of the American Museum of Safety in New York, was a report from the General Electric Co. on safeguarding machine tools. The company has made a careful study of this subject and seventy-five to eighty per cent of its 10,000 machine tools now have safety devices, the remainder being fitted out as rapidly as possible. A committee on safety devices has been organized with representatives from each of the principal works, who meet periodically to discuss and adopt standard protective devices. When this committee has agreed on standard devices it becomes the duty of the local committees of each plant to see that they are applied and kept in working condition.

While sentiment and altruistic impulse may play some part in the progress of industrial safety, business policy is the prime moving force. Corporations, large and small, are waking to a realization of the fact that dangerous conditions should not be tolerated in their plants. Systematic study of accidents has shown that simple precautions will eliminate a large percentage of them. Workmen are also beginning to realize more clearly their duty to themselves and their fellows to avoid the causes of accident and to regard and utilize the means provided to prevent them. This is indicated by the great improvement made by such concerns as the United States Steel Corporation, which employs 200,000 workmen, and which has reduced its accident roll forty-five per cent. The Midvale Steel Co. has cut down its accident list sixty-one per cent and the Pennsylvania R. R. Co. has reduced accidents to its 33,000 employes in the shops, sixty-three per cent. It is evident that the efforts of employers to promote safety are appreciated and that the spirit of cooperation has been awakened.

* * *

INTERNAL COMBUSTION ENGINE FUELS

The large number of automobiles and other motors now in use which consume gasoline has tremendously increased the consumption of this fuel during the past ten years. Not many years ago gasoline was an almost useless by-product, disposed of with difficulty by the oil refiners, but as the internal combustion engine became the leading vehicle motor, the demand increased so greatly that kerosene is now the by-product and gasoline the prized product. The high price of gasoline is a serious matter to makers and users of automobiles, and the need of engines that can use kerosene is imperative. The promise of alcohol as a cheap motor fuel has not been realized. The high prices of grains and vegetables from which alcohol can be produced have prevented it, as have also the government restrictions which make the operation of small plants impracticable.

Kerosene is superior to gasoline as regards safety, price and availability, but its high gasifying temperature which makes it comparatively safe to handle, is a detriment as an internal combustion engine fuel. It is true, of course, that internal combustion engines using kerosene have been in use for years; but they are not so flexible nor so easily started as gasoline motors. Various efforts have been made to adapt kerosene as a substitute for gasoline, and it seems probable that the substitution will be accomplished for automobiles in the near future. One plan is to provide two carbureters and two fuel reservoirs, one for kerosene and the other for gasoline, and when the engine is well warmed up to turn on the kerosene supply and shut off the gasoline. Each fuel being led through its own carbureter, no change of adjustment is necessary. In winter it is desirable to heat the kerosene supply, and that is readily accomplished by diverting the exhaust into a heating coil surrounding the kerosene tank.

THE MECHANICAL ENGINEER AND THE PANAMA CANAL

The leading article in this number of MACHINERY describes some of the mechanical engineering problems that have been solved in the carrying out of one of the greatest engineering undertakings of the world. It is interesting to note the part that the mechanical and electrical engineers have played in this great development. In general the visitor approaches Panama with the idea that the Canal is merely a great civil engineering undertaking; but when he sees the work in progress, he cannot fail to realize that tremendous as is the work of the civil engineer, many of the salient features of this wonderful connecting link between the two great oceans are purely mechanical in their character. The huge lock-gates and culvert valves with their operating mechanism, as well as the ingenious interlocking system which makes for absolute safety, are monuments to the skill and inventive ability of mechanical and electrical engineers.

The time has passed when the civil engineer, as such, can alone carry out an engineering work of the magnitude here met with. While he is in complete charge and must have a perspective of the whole work he depends to a very great extent upon other branches of engineering. All the machinery used in the excavation of the canal is the product of the mechanical engineer; and the operation of the canal, when once completed, will largely make use of electrical apparatus. It is doubtful if there is any engineering undertaking that has required the cooperation of so many different branches of engineering as has the Panama Canal, and it is therefore fitting that a mechanical journal such as MACHINERY should give an adequate description of some of the mechanical features of the construction.

* * *

THE NEED OF STANDARDIZATION

The need of standardization and the service which the Engineering Standards Committee of Great Britain has done in the fields of standardization which it has taken up, is well illustrated by the following incident: A few weeks ago a correspondent wrote to us complaining that a table of Whitworth standard pipe threads, as published in one of MACHINERY'S reference books, did not agree with data given in a table published in 1901. Upon investigating the matter and noting the sources from which the published information had been obtained, it was found that the table published in MACHINERY'S reference book on screw threads, which was based upon the practice of one of the most prominent tap manufacturers in the country, did not agree with that of another well-known manufacturer of these tools, and the practice of neither of these manufacturers agreed with the table published in 1901. All three of these tables, however, represented "standards" of manufacturers of high reputation in their field. Of course, neither of the three "standards" agreed with the newly accepted standard of the British Engineering Committee, although its table published in 1901 gave dimensions very close to those now accepted. In another case, a smaller tool manufacturing concern, working to the figures published in the catalogue of one of the leading makers, had a large number of taps rejected because they were not "standard."

It is difficult for an individual firm to procure the adoption of a standard, but an engineering society of established standing could have taken this work up long ago, and the confusion that now exists would have been avoided. In a few years we hope that in the practice referred to the recommendations of the British Engineering Committee will be adopted by all manufacturers; but notwithstanding the fact that the report of this committee was published in 1905, we find the old dimensions given by manufacturers in some of the small-tool catalogues of 1912.

There are scores of other instances where standardization would be a great advantage to both manufacturers and users. The engineering societies still have a great work ahead of them—work that will mean a great deal more to the engineering world than merely coming together at annual meetings and reading papers which are often received with indifferent interest.

ON SUGGESTION SYSTEMS

BY F. W. HARRIS*

It has always seemed to me that the suggestion box idea was a good one but I have been experiencing some trouble with such a scheme, and it may be interesting to see if others have encountered the same difficulty. We started with the assumption that our machinists were working largely with their hands but that there was no reason why they should not have an occasional good idea and several hundred idle brains might just as well be set to work for the benefit of the concern.

We installed boxes and offered prizes for the best suggestions and after several months are now taking stock. It does not seem to us that the results have been profitable. We have managed to get several very good suggestions that have more than paid us for our trouble, but we have apparently sowed the seeds of discord and made several very discontented men who are growling about the unfairness of employers in general and of us in particular. This is causing a loss in efficiency that seems to me to more than balance the gain that we have made. We wanted to encourage our men to think, and to show them that we appreciated their giving us the benefit of their thoughts.

The trouble seems to be that nine-tenths of the suggestions made are of no value and the suggestors will not take a turn-down without getting "sore". Bill Jones, working on a lathe, gets an idea in his noddle that the design of a particular part is not right, and he starts spending his evenings improving it. He will not take any one into his confidence as he does not want his ideas stolen, and the result is that after a lot of trouble he turns in a suggestion that is entirely impracticable. Anyone who has had experience with a lot of engineers fighting about design knows how fond one grows of his own notions along such lines, and can appreciate how hard it is to show Bill that his pet scheme is utterly worthless. We tried it and gave it up; if the scheme is good—which happens one time in ten—we use it, but if it isn't we send Bill a very neatly printed note saying that his scheme is a good one but that we cannot use it—because of certain reasons which we cannot explain. We then go on to say that there are many reasons quite aside from lack of merit that compel us to turn down some schemes, such as patents that we dare not infringe, etc., and wind up by asking Bill to try again.

Bill does try again and he gets another slip reading just the same. With only one idea in ten proving worth, while there are bound to be a lot of blanks, and every blank is regarded as more or less of an injustice. Then there has been some conflict of ideas; men often discuss their schemes and several times two men have put in the same idea. It has proved a hard matter to find out just who originated it. I have about decided that when you start to hold out promises of rewards for such matters you are more or less bound to stir up a hornet's nest.

Then there is the fellow that has made a suggestion to his boss, and has been turned down. He slipped the same scheme in the box and pretty soon the general foreman asked the foreman about the scheme. The foreman knew right away that the workman had gone over his head and that did not help to make relations pleasant. During our several months' use of the suggestion box a score of schemes have turned up that have been taken up and discussed and discarded after giving as thorough a consideration as practicable. On tracing the matter we have found that these were brought up by the same men who originally started them and who felt that their ideas were not adopted because of some trickery or bias. These incidents not only have caused a lot of extra work but some hard feeling. I personally have about decided that we should discontinue the boxes, but it would be interesting to hear from others who have tried them.

* * *

To find the weight of a casting when the weight of the wooden pattern can be ascertained, multiply the weight of the pattern by 9 if it is made of oak; by 9.7 if it is made of beech; by 12.8 if it is made of birch; by 13.4 if it is made of linden, and by 14 if it is made of pine.

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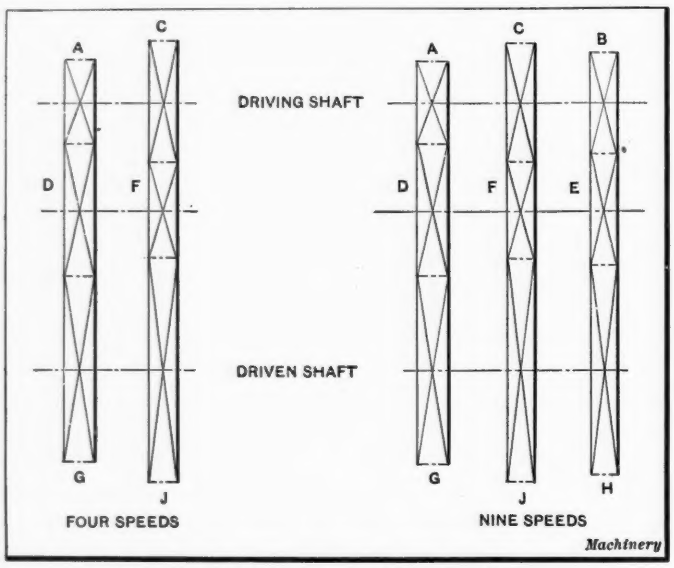
FIGURING OUT GEARS FOR SPEED-BOXES

BY J. J. DENSMAN

Commonly used arrangements of speed-box gears giving four and nine speeds, respectively, are shown in the accompanying illustration. The gears on the driving and driven shafts are sometimes arranged to slide into engagement with those on the intermediate shaft; such gears are also designed to be driven or to drive through friction clutches, tooth clutches or sliding spring keys. In this type of speed-box, it is impossible to select gears so that the speeds of the driven shaft are in strict geometrical progression, although it has



Gear Boxes which give Four and Nine Speed Changes

been stated that the speeds of certain machine tools in which this type of box is employed are in geometrical progression.

The writer has not come across any published information relating to the method of figuring out these boxes, and believes that the usual practice is to "cut-and-try" until satisfactory results are finally obtained. This formerly made it necessary for some hours to be spent in working out a box before arriving at a passable result, and there was then an uncertainty as to whether or not the best combination had been found. Consequently, the writer developed the following formulas which have been the means of effecting a great saving of time. These formulas do not lead directly to a complete solution, as there are many combinations of gears which will give the limiting speeds, but they narrow the choice of the dimensions for certain gears between fixed limits.

As a rule, such boxes are driven by a high-speed pulley or by a motor, and the ratio of the driving shaft speed to the minimum driven shaft speed is greater than that of the maximum driven shaft speed to the driving shaft speed. This leads to the selection of gear A on the driving shaft as the gear having the smallest number of teeth. For simplicity in dealing with the formulas, the number of teeth on this gear will be taken as unity. Let

S = speed of driving shaft in R. P. M.

X = maximum speed of driven shaft in R. P. M.

Y = minimum speed of driven shaft in R. P. M.

Then using notation shown in the illustration

$$J \text{ must not exceed } \frac{S}{Y} \tag{1}$$

and must be of such a value that

$$J\left(J^2 + J - \frac{S}{Y}\right) > \frac{S^3}{XY^2} \tag{2}$$

From these relations we obtain limiting values of J.

Then

$$C = \frac{J+1}{\dfrac{S^2}{XYJ} + 1} \tag{3}$$

and

$$F = \frac{YJ\,(C-1)}{S - YJ} \tag{4}$$

Having found values for J, C and F, it is a simple matter to obtain those for D and G from the equations

$$C - A = J - G$$
$$C + F = A + D$$

A good trial value for B in the nine-speed box is

$$B = \frac{J - C}{\sqrt{\dfrac{JG}{C} - 1}} \tag{5}$$

from which we obtain values for E and H.

By deciding on a certain number of teeth for gear A, we obtain numbers of teeth for the rest of the gears from the product of this number and the respective values obtained from the preceding formulas, the results being taken to the nearest integers. After a few trials with various values of J between the limits mentioned—the tendency of the resulting speeds being noted—the best arrangement is quickly obtained with the knowledge that it is the best for the number of teeth assumed for the gear A. By the best arrangement is meant the one whereby the speeds obtained are nearest to a geometrical progression. Some of the formulas may appear a little complicated but with the aid of a slide-rule they are quickly worked out. It might be mentioned that with values of J approaching the high limiting value, the values of F are high, becoming infinite in the limiting case, and are near to unity as the values of J are taken close to the minimum.

It may be of interest to show how these formulas were obtained.

$$X = S\frac{CD}{FG}; \qquad \frac{F}{D} = \frac{SC}{XG} \tag{6}$$

$$Y = S\frac{AF}{DJ}\ (A = \text{unity}); \qquad \frac{F}{D} = \frac{YJ}{S} \tag{7}$$

From (6) and (7)

$$\frac{SC}{XG} = \frac{YJ}{S}$$
$$S^2C = XYJG$$

and as

$$G = J - C + 1, \quad S^2C = XYJ^2 - XYJC + XYJ \text{ and}$$

$$C = \frac{J+1}{\dfrac{S^2}{XYJ} + 1} \tag{3}$$

From (7)

$$\frac{F}{D} = \frac{YJ}{S}$$

and as

$$D = C + F - 1$$
$$FS = YJC + YJF - YJ$$

and

$$F\,(S - YJ) = YJ\,(C - 1) \tag{4}$$

$$F = \frac{YJ\,(C-1)}{S - YJ}$$

Now for F to be greater than unity it is necessary that

$$S - YJ < YJ\,(C - 1)$$

That is

$$S < YJC.$$

Substituting the value of C from (3) and simplifying, we get

$$J\left(J^2 + J - \frac{S}{Y}\right) > \frac{S^3}{XY^2} \tag{2}$$

From (4), S must exceed YJ for F to have a positive value.

$$\text{Therefore } J \text{ must not exceed } \frac{S}{Y}. \tag{1}$$

B is found by assuming $\frac{A}{G}$, $\frac{B}{H}$ and $\frac{C}{J}$ to be in geometrical

progression. In using the formulas, it will be found advantageous to work out the values of $\frac{S}{Y}$, $\frac{S^2}{XY}$ and $\frac{S^3}{XY^2}$ separately for use in Equations (1), (3) and (2) respectively, with the different trial values of J, etc.

To illustrate the use of this method of figuring the gears for speed-boxes, we will assume a case in which the following speeds are required:

Driving shaft speed = 300 R. P. M. = S
Maximum driven shaft speed = 720 R. P. M. = X
Minimum driven shaft speed = 63 R. P. M. = Y

Then J cannot exceed $\frac{300}{63} = 4.76$, and a minimum value for J is obtained from $J\left(J^2 + J - 4.76\right) > \frac{300^3}{720 \times 63^2}$. From this data a value of 2.48 is obtained for J.

Commencing with 2.5 as the value for J

$$C = \frac{J+1}{\frac{S^2}{XYJ} + 1} = \frac{2.5 + 1}{\frac{300^2}{720 \times 63 \times 2.5} + 1} = 1.95$$

$$F = \frac{YJ\,(C-1)}{S - YJ} = \frac{63 \times 2.5\,(1.95 - 1)}{300 - 63 \times 2.5} = 1.06$$

Values for D and G can now be obtained by inspection.

$D = 2.01$
$G = 1.55$

$$B = \frac{J - C}{\sqrt{\frac{JG}{C} - 1}} = \frac{2.50 - 1.95}{\sqrt{\frac{2.5 \times 1.55}{1.95} - 1}} = 1.34$$

We then obtain values for E and H by inspection

$E = 1.67$ $H = 1.89$

Now assume that the gear A has 20 teeth. The numbers of teeth in the other gears will then be obtained by multiplying the preceding results by 20. Taking the results to the nearest integers we have:

$A = 20$ teeth, $B = 27$ teeth, $C = 39$ teeth, $D = 40$ teeth, $E = 33$ teeth, $F = 21$ teeth, $G = 31$ teeth, $H = 38$ teeth, $J = 50$ teeth.

The available speeds of the third shaft from this set are thus: 63, 103, 130, 194, 213, 234, 317, 484, and 719.

Other values of J may be tried and the most suitable set of speeds decided upon. It will be noticed that higher values for J will make the middle shaft gears larger and larger until they approach infinity for sizes nearing the maximum value of J. For some ranges of speed, values for B have to be modified as the speeds "cross" each other. It seems hardly necessary to mention the advantage of the slide-rule in dealing with these formulas.

* * *

COMBINATION OF NON-COMPETITORS UNDER THE SHERMAN LAW

In handing down the decision of the United States Supreme Court in the case of the Government against the United States Shoe Machinery Co., charged with violation of the Sherman Anti-Trust law, Justice Holmes stated that there can be no greater objection to one corporation manufacturing 70 per cent of three non-competing groups of patented machines collectively used for making a single product, than to three corporations making the same proportion of one group each. The disintegration aimed at by the statute does not extend to reducing all manufacture to isolated units of the lowest degree. As there had been no claim by the Government of unfair competition, the unanimous decision of the court held that on the face of it the combination was simply an effort for greater efficiency. The business of the several groups that combined was assumed to have been legal before the combination, and that being the case, the court states that it is difficult to see why the collective business should be any worse than its component parts. The court thus clearly declares that the Sherman Anti-Trust law does not forbid the mere combining of non-competitors in an industry.

PUNCHING EFFORT*

In punch work, there are a number of influences which affect the effort necessary to punch a certain sized hole through a given thickness. Principal among these are the relative proportions of the punch and die, for if the punch fit the die snugly, not only is there shearing action to be overcome, but in addition, a considerable binding of the punching in the punched plate and die. It has been found that the effort required is materially increased when the fit is snug, this also tending to shorten the life of the punch. If, however, the size of the hole be slightly increased, within certain limits the punching effort will be decreased, and the life of the die lengthened. Practice shows that for plate work, if the die be 1/32 inch larger than the punch, giving 1/64 inch clearance all around, conditions are much improved. However, if this amount be increased too much, the resulting punched hole will be conical in shape, and consequently, it is inadvisable to increase the clearance above the amount stated.

The punch and die are both given clearance to facilitate stripping the punched plate and punching, respectively. In the die, this clearance is given in various ways. In some cases, the inner wall is given a slight batter, the diameter at the cutting edge being as stated, and increasing slightly from this point down. In other cases, the wall directly below the cutting edge is left parallel for possibly ⅛ inch, with the hole enlarged for the remaining part of the die. Similarly, the punch is made either tapering gradually back from the cutting edge, or else receding sharply to a predetermined body size of uniform diameter. In bridge work the latter is more generally used, and is preferred, for the reason that when the cutting edges are dulled, it does not bind like the uniform taper punch.

In the Data Sheet Supplement it is assumed that the punch and die are so proportioned that the shearing effort alone need be considered, neglecting the previously mentioned binding action. In it are tabulated the pressures required to punch 0.25 carbon steel of a tensile strength of 65,000 pounds per square inch. It constitutes a very convenient reference table, making unnecessary much of the routine work usually accompanying such problems.

The effort required to perform this work is found by obtaining the product of the circumference of the hole by the thickness of the plate, which represents the area under shear, and multiplying this by the ultimate shearing strength of the plate. That is

$$P = \pi d t f_s$$

in which

P = punching effort or pressure,
d = diameter of hole,
t = thickness of plate, and
f_s = ultimate strength in shear.

For an example, consider a 1⅛ inch hole in a ¾ inch plate, where the shearing strength of the latter is 50,000 as on the Data Sheet. Then

$$P = 3.1416 \times 1.125 \times 0.75 \times 50,000$$
$$= 132,500 \text{ pounds.}$$

The employment of the table thus saves much work.

Punching, while not the best method of making holes in plates or structural shapes, is in almost universal use, with the possible exception of heavy marine boiler work, in which case it is customary to drill the rivet holes. The drilling process is preferred in that case, as punching has a tendency to crystallize the metal around the hole and weaken it, a condition that might prove fatal in large work which is more subject to this action than thinner plates.

* * *

A committee has been appointed by the British Government to inquire into and report on the conditions necessary for the adequate and suitable lighting of factories and work shops, with special reference to the nature of the work which is carried on and the protection of the eye-sight of the persons employed.

* With Data Sheet Supplement.

LOCATING THE DRIVING SCREW ON HORIZONTAL BORING MILLS

BY W. G. DUNKLEY*

In a recent article on the location of supporting chains for the spindle slide of horizontal boring machines, the writer discussed the question of the turning moment on the spindle slide in the plane of the upright face under various conditions. It was also mentioned that there may be a turning moment in a plane at right angles to the upright, which would cause a binding action between the slide and upright in this plane. Assuming the same loads as were used in the previous article—a slide weighing 8 tons, and balance weight weighing 4 tons—if the conditions are as shown in Fig. 1, the upward reaction of the screw on the slide and the total pull in the chains will be equal. The resultant of these two forces of 4 tons will be a force of 8 tons acting midway between their points of application along the line *D–D*, and the

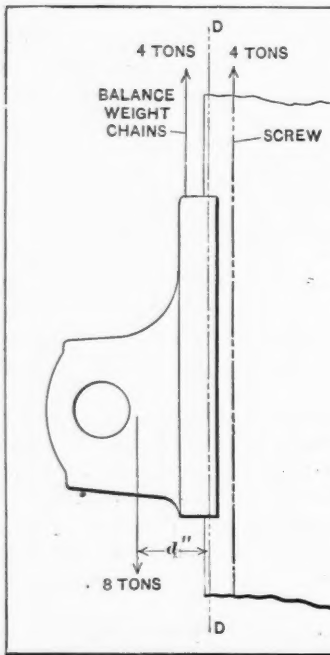


Fig. 1. Location of Resultant of Forces on Slide

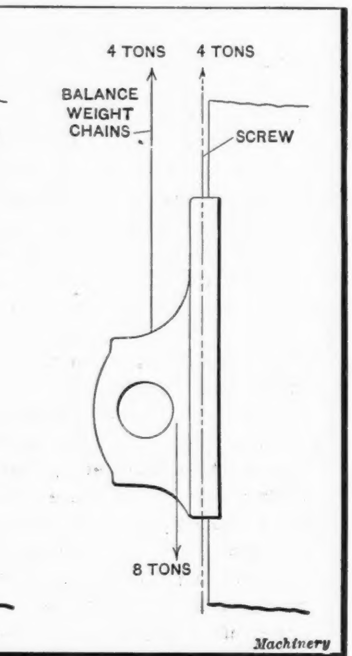


Fig. 2. Conditions which make Twisting Moment Zero

turning moment on the slide will be $8d$ ton-inches. A consideration of the illustration will show that it is desirable to reduce the distance d as much as possible. This is done by keeping the center of gravity of the slide as near to the upright as possible and also by bringing the screw out as near as possible to the center of gravity of the slide in this plane. The chains should be secured on the opposite side of the center of gravity from the screw, their location being brought as nearly as possible to meet the conditions in Fig. 2, where the twisting moment on the slide is zero.

In the article referred to, the writer also pointed out that it was desirable to keep the vertical screw as near as possible to the face of the cut in the plane of the face of the upright, where the machine was engaged on vertical milling. Provided the supporting chains are properly placed with respect to the position of the vertical screw and the center of gravity of the slide, it was shown that this arrangement reduced the turning moment in this plane to a minimum when the slide was stationary. The question now arises, "Does this also apply to the horizontal driving screw in the bed?" It is never safe to assume that because a calculation holds good for one case it will hold good for another case. Bearing this in mind, it will be interesting to look into the influence which the position of the horizontal driving screw in the bed exerts upon the turning moment on the upright. As an example, let us consider the case in which the center of gravity of the load on

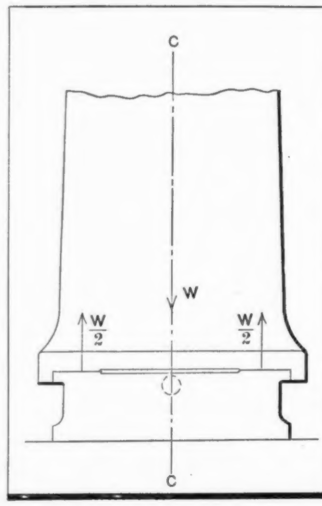


Fig. 3. Distribution of Load on Bed

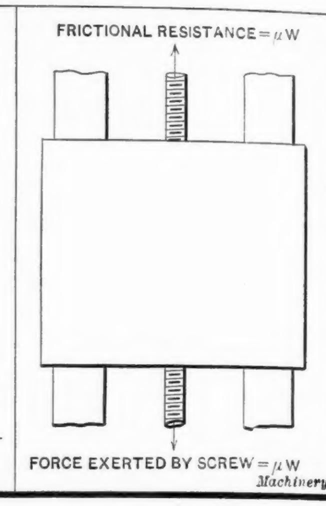


Fig. 4. Amount of Frictional Resistance to Motion

the bed—consisting of the weight of the upright, spindle slide and balance weight—is located on line *C–C* in Fig. 3, which passes through the middle of the bed if the bed is equally loaded on the two surfaces. Denoting the load by W tons, the resistance to motion due to frictional forces on the top of the bed will be μW tons, where μ is the coefficient of friction. The resultant of this frictional resistance will come directly beneath the center of gravity, and assuming that the screw is located under the line *C–C*, which is in the middle of the bed in this case, the force on the screw will be equal to μW in order to just cause the upright to move. The turning moment of the upright will be zero as shown in Fig. 4.

Assuming a case where the machine is milling horizontally and the resistance of the cut is P tons acting at a distance L from the middle of the bed, there will be a turning moment giving rise to the forces Q on the edges of the bed. These forces Q will cause additional frictional resistance μQ along each edge of the bed, as shown in Fig. 5. For these conditions, the screw has to overcome the force P due to the resistance of the cut and the frictional resistance $\mu W + 2\mu Q$. This makes the total force between the screw and upright to just cause motion equal $P + \mu W + 2\mu Q$. Taking moments of the forces parallel to the screw about an axis through the screw, we get a turning moment of

$$PL + \mu Qx - \mu Qx = PL.$$

This moment is balanced by the forces Q from which we find

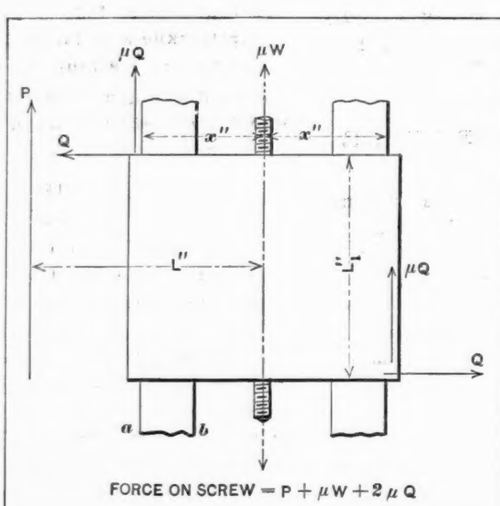


Fig. 5. Frictional Resistance due to Turning Moment

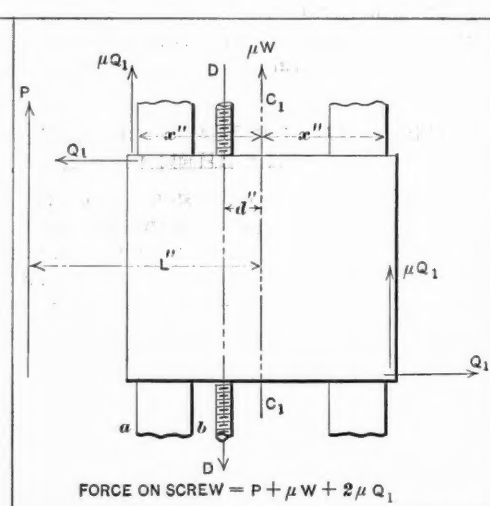


Fig. 6. Effect of varying Position of Driving Screw

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$QL_1 = PL$. The transverse frictional resistances on the top of the bed would reduce the values of the forces Q in this case, but this fact will not affect the deductions arrived at with respect to the position of the screw, as we are not so much concerned with the forces resisting the turning moment as with the forces which tend to augment it. Thus when the screw is under the center of gravity, we have no turning moment on the upright when it is being traversed, but there is a turning moment of PL ton-inches when the machine is engaged in milling.

Let us next consider a case in which the screw is moved a distance d toward the point of application of the force P, as shown in Fig. 6. Taking moments about the middle of the bed, the turning moment of all forces parallel with the screw will be zero when the machine is engaged in milling. This condition is expressed in the following equation:

$$PL + \mu Q_1 x - \mu Q_1 x - (P + \mu W + 2\mu Q)\, d = 0$$

from which we find that the value of d which will give a zero turning moment is:

$$d = \frac{Pl}{P + \mu W + 2\mu Q_1}$$

But for such a case, we would have a turning moment equal to

$$(\mu W + 2\mu Q_2)\, d$$

when the machine is being traversed. In this case, W equals 25 tons, so that the turning moment is considerable, and in determining the best position for the horizontal screw, we have to choose between two evils: shall we arrange to have the turning moment a minimum when the machine is engaged in milling, or when the upright is traversed over the bed? Evidently we cannot secure the advantages of the best location for each individual condition. The best we can do is to have one-half the turning moment in each case by putting the screw somewhere between the lines D-D and C_1-C_1. It is evidently poor designing to put the screw to the right of the lines C_1-C_1, as this brings it further from the point of application of the force P and makes the turning moment greater for both milling and traversing.

The question of the narrow guide does not properly come within the scope of this article but it seems advisable to point out that with the guide which is at a and b as shown in Fig. 5, the only alteration would be that the forces Q would act along a and b instead of in the manner shown. The result in this instance would be an increase in the turning moment due to the forces parallel to the screw which shows a slight disadvantage in this respect for the narrow guide as compared with the broad guide.

It is obvious that no golden rule can be laid down respecting the position of the driving screw of the machine, since it can only be decided from a consideration of the circumstances which govern individual cases. It appears that little improvement in the amount of turning moment can be obtained by using two driving screws instead of one. Referring to Fig. 7, we will assume that the forces on the two driving screws are equal, the force in each case being $\frac{K}{2}$, in which K is the resultant of all resisting forces, its point of application being at a distance L from the line D-D midway between the center of the two screws. Taking moments about D-D, the turning moment is $KL + \frac{Kx}{2} - \frac{Kx}{2} = KL$. In Fig. 8, a single driving screw C is substituted for the two screws of the preceding illustration. The force on the screw in this case will be K and the turning moment KL as before, from which it will be seen that, so far as the turning moment is concerned, there seems to be little advantage rising from the use of two screws. Of course, the pressure on each of the two screws is less than where single screws are used, with the result that there is less wear. There may also be other points of advantage which have no direct bearing upon the subject of this article but which might have to be considered when deciding whether two screws or one are best suited for the requirements of the machine that is being designed.

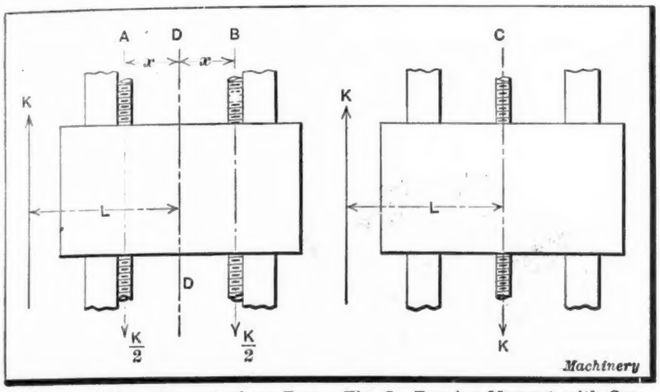


Fig. 7. Turning Moment where Two Driving Screws are used Fig. 8. Turning Moment with One Screw instead of Two

* * *

WINDING PIANO WIRE TENSION SPRINGS*

BY C. R. T.

Most mechanics have their own method of winding springs, but many of these do not comply with the most approved practice. During the writer's experience as a mechanic, he has had occasion to wind practically every type of tension spring, and the information which he has gathered on this subject is presented in tables which appear in the current Data Sheet Supplement. These tables are practically self-explanatory, but it may not be out of place to cite typical examples showing the method of using them:

For this purpose it will be assumed that a mechanic is required to wind three different springs; the first to be wound with 0.035-inch wire and to fit in a ¾-inch hole; the second to be wound with 0.040-inch wire and to fit in a ⅜-inch hole; the third to be wound with 0.060-inch wire and to be a sliding fit on a ½-inch shaft. Reference to the tables will show the proper sizes of mandrels to be as follows: For the first spring, 0.625 inch; the second spring, 0.250 inch; and the third spring, 0.437 inch, 0.011 inch being allowed for play between the spring and shaft in the latter case.

In all cases where the mandrel diameter is in excess of ⅜ inch, the mandrel is mounted in a lathe chuck. Mandrels of less than ⅜-inch diameter are mounted in a drill chuck. In fastening the wire in a lathe chuck one jaw is usually loosened and where the mandrel is driven by a drill chuck, the wire is placed between the jaws and the mandrel. If a long spring is required, a mandrel of corresponding length is used, which is ground to an angle of 60 degrees at the end to fit into a female dead center for support. The wire is placed in a bench lathe boring-tool holder or a V-holder in the toolpost. A piece of brass about ⅛ by ½ by 3 inches is placed between the wire and the toolpost screw. A groove 1/32 of an inch deep is filed in this brass to hold the wire in place. The groove is filed in the lengthwise direction of the brass plate and is made of the proper depth for the size of wire from which the springs are being wound. This clamping arrangement is tightened up with the toolpost wrench, just enough tension being put on the wrench to keep the wire from slipping.

When it is required to wind springs of a standard pitch, provision is made for this purpose by means of the lathe gears, the pitch of standard springs being the same as that of standard screws of corresponding diameters. Consequently the customary tables used for screw cutting can also be used for winding springs according to this method.

* * *

Aluminum is being used to some extent for jig and fixture frames. It has the advantage of being only about one-third as heavy as cast iron, which is the reason, of course, for using it. Machine operators can readily handle an aluminum jig which would be beyond their strength if made of cast iron. The cost of aluminum jigs is higher than that of cast-iron jigs, but not in proportion to the relative costs per pound. Aluminum weighs, bulk for bulk, only about one-third as much as cast iron, and at the present market price the bulk cost is about the same as cast iron at from seven to eight cents a pound.

* With Data Sheet Supplement.

MAKING STEEL TYPE*

CUTTING LETTERS IN STEEL BY HAND AND MACHINE

BY CHESTER L. LUCAS†

The general method of making type of type metal (tin, lead and antimony) for printing is well known, but to make type from steel so that it can be set up and used for stamping metal in the same way that printing type is used, is quite another matter. The Schoder & Lombard Engraving Co., 18 Franklin St., New York City, makes quantities of steel type that can be set and locked in chases the same as regular printing type, and some of the methods used in the manufacture should be interesting to those not familiar with this class of work. In making steel type the perfect alignment, even length, uniformity of face, etc., are as necessary as for printing type. Steel, of course, cannot be cast in type form; therefore, the letters must be cut upon the type face starting from the solid steel. This type is made for stamping steel, brass and wooden novelties with makers' names, etc., and also for printing on cloth and other materials where ordinary type would not give long service. For such purposes the type can be set up and used exactly the same as ordinary type. At *A* and *B*, Fig. 1, are shown pieces stamped with this type and in Fig. 1 separate types may be seen at *E*, *F* and *G*.

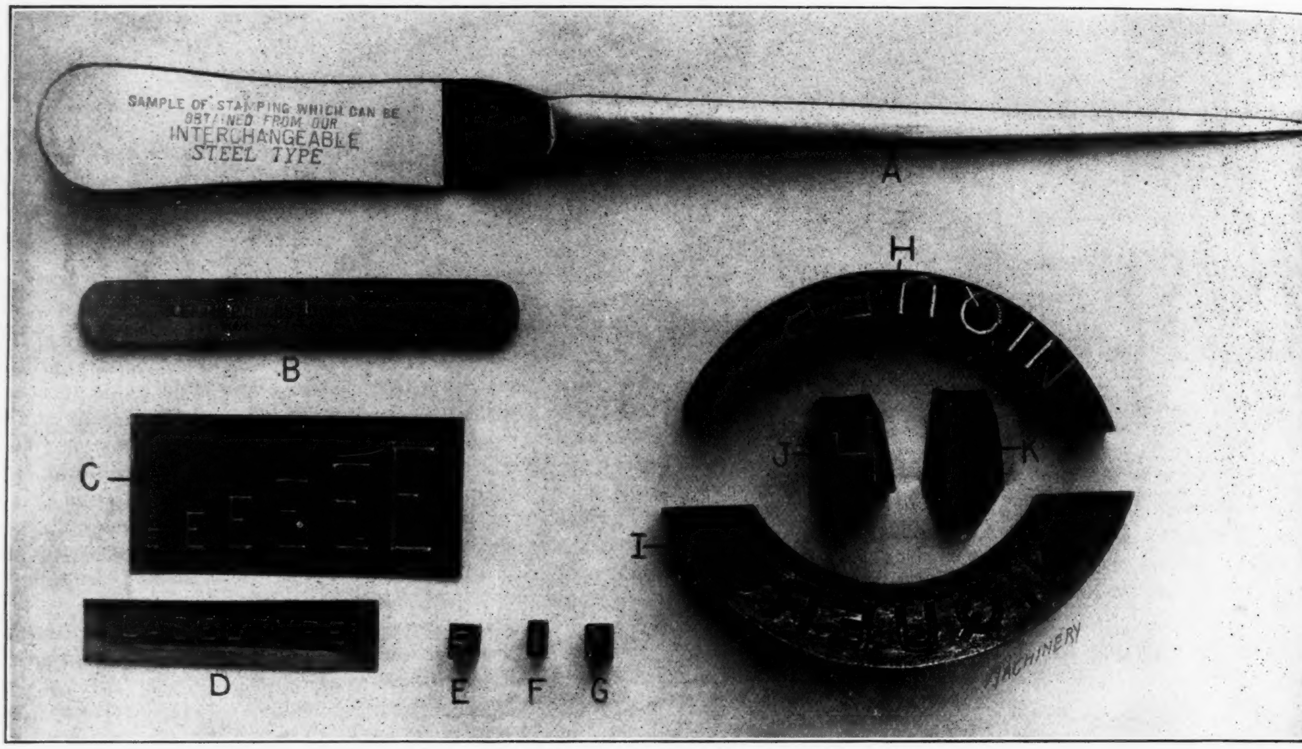


Fig. 1. Samples of Steel Type and Work stamped from them

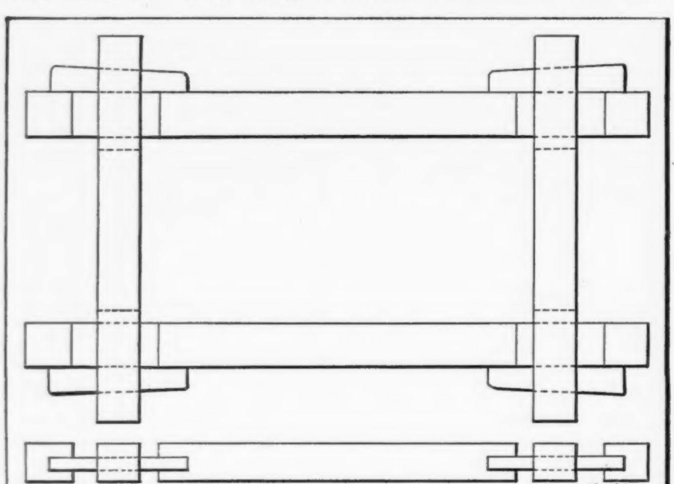


Fig. 2. The Chase in which the Type is held for Grinding

Preparing the Blanks

The stock from which steel type is made is cold-drawn high carbon steel, and the three-foot long strips come drawn very accurately to size. In making a given font, stock is selected of such proportion that all the letters may be made upon a standard size of blank excepting the letter I which requires a narrow blank. By making a special style of M and W, the use of wide blanks is obviated. As the type is finished to a length of 0.625 inch (standard printing type is 0.918 inch high), the blanks are sawed to this length plus 0.005 inch—just enough to allow for finishing. A 5A font has 150 characters; therefore this number of type blanks is cut off, using a cutting-off fixture on a milling machine, while clamped in the special chase shown in Fig. 2. This chase is a simple outline with adjustable sides which may be keyed about the font of the type. Different sizes of these chases are used to accommodate fonts of large or small type. Keyed in this frame the type blanks are taken to a surface grinder and while held on a magnetic chuck they are ground to a length of exactly 0.625 inch. In order to keep this size uniform, an ordinary limit gage is used for checking the lengths of the blanks. This gage is shown at *A*, Fig. 12.

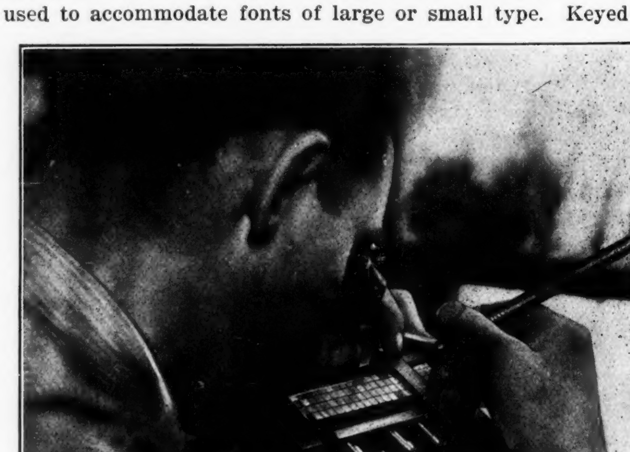


Fig. 3. Laying out the Letters on the Type

*For further information on cutting letters in steel, see "Steel Letter Stamping Dies," published in MACHINERY, January, 1912, engineering edition.
† Associate Editor of MACHINERY.

Routing out the Type

The blanks, which are now all of standard length, are next locked in a special holder shown in use in Fig. 3 which is used for handling the type from this stage until ready for hardening. The blanks are set up in this box with thin spaces separating them and strips to separate the rows. The object of separating the blanks will be made apparent later. By tightening the set-screws on the sides and end of this holder the blanks are held rigidly so that they can be machined without difficulty. While locked in this box any necessary laying out is done as shown in Fig. 3. Some of the cutting may be done without previous laying out, such letters as O, U and other easily made letters are handled in this manner. The more difficult shaped characters like R, S, and & are usually carefully laid out before cutting.

One of the machines used for routing or milling out is illustrated in Fig. 4. This is a Becker vertical milling machine specially adapted for this class of work. The box with the blanks locked in it is clamped on the table of the machine, and by means of special high-speed cutters the letters are routed out. At A in Fig. 5 is shown the style of cutter used almost exclusively on this work. These cutters are made of high-speed steel shaped like a diamond, as indicated in the illustration. It is obvious that the two lips of the diamond do the cutting and as the point of the cutter is sharpened chisel-shape, it may be fed downward when reaching into the center of a letter O or similar character. These cutters are finished with an average included angle of 30 degrees.

Fig. 4. The Milling Machine on which the Type is routed

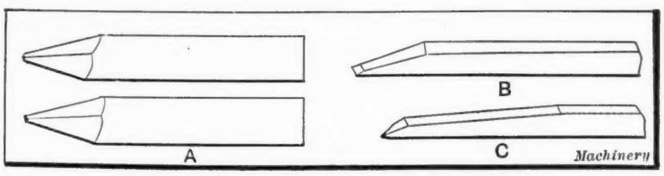


Fig. 5. Cutter and Gravers used in Type Making

If the letters must be cut deep and narrow, it is necessary to use a cutter of less degree. At best, this work is hard on the cutters and they must be favored as much as possible in order to keep them in good condition. It is much easier to start a small cutter like this from the side of the work than to force it to cut to its depth straight downward; and it is for this reason that the blanks are separated by a space in the holder. In routing out the different parts of the work the cutter is guided by both of the feed handles. The handles are worked separately over the horizontal and perpendicular lines and simultaneously over the inclined lines. Through continued practice an operator becomes adept in its manipulation. An air blast is provided to keep the chips clear from the cutting point, and the work is done without the use of lubricant of any kind. Large cutters are used wherever possible, for they, of course, give the best service. The average cutter speed is 1400 R. P. M. To keep the cutting to standard depth, the adjustable stop for the spindle is set, and the cutter is brought down to the right depth by pressure upon a foot treadle, thus leaving both hands free for the operation of the feed handles.

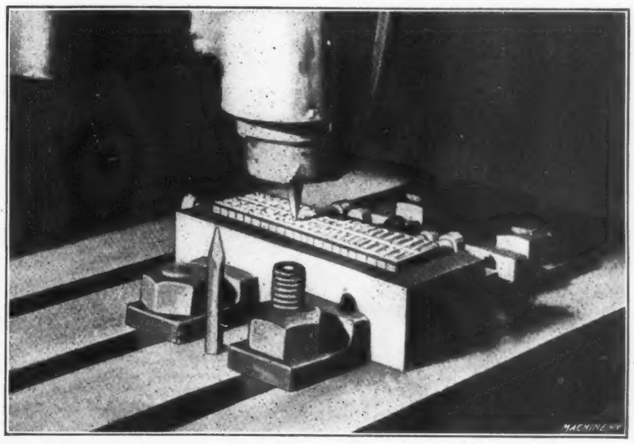


Fig. 6. Detail View showing how the Cutter works on the Type

Hand chipping or filing is not resorted to. By relying solely upon machine work for removing the metal several advantages are gained. In the first place the bevels are kept uniform, which would be practically impossible if a file or chisel were used. Even bevels mean that no matter how deep the type is forced into the material to be stamped the impression will be even. By the use of the machine the horizontal and perpendicular lines are kept at right angles to each other so that at the final finishing it is only necessary to smooth up the sides, which can easily be done without losing the alignment of the letter. The third reason for milling the entire letter rather than filing or chipping the outside is that the work may be finished completely in the holder, thus requiring a minimum of handling in producing the work. Referring to Fig. 6 it will be seen just how the cutter is used in routing out the type.

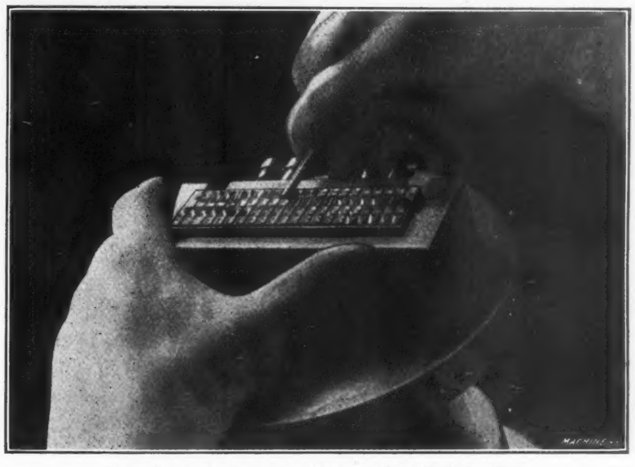


Fig. 7. Trimming the Type with a Flat Graver

Engraving the Type

There is, of course, a limit to the amount of work which can be done on the blanks by routing. After this point has been reached the remainder of the work must be accomplished by hand, using engraving tools like those shown at B and C, Fig. 5, for the purpose. Fig. 7 illustrates a type-cutter starting to trim up a font of type. Here he is shown with a flat graver squaring up the sides of the routed letters, an opera-

tion which requires a skilled workman. For this hand work the type is still retained in the holder, being mounted in the engraver's vise shown in this illustration. The tops and bottoms of the letters are first squared up, then the outsides of the letters are similarly treated with the flat graver which is shown in detail at *B* in Fig. 5. Next the pointed graver, illustrated at *C*, Fig. 5, and known to the trade as a "Spitz tool,"

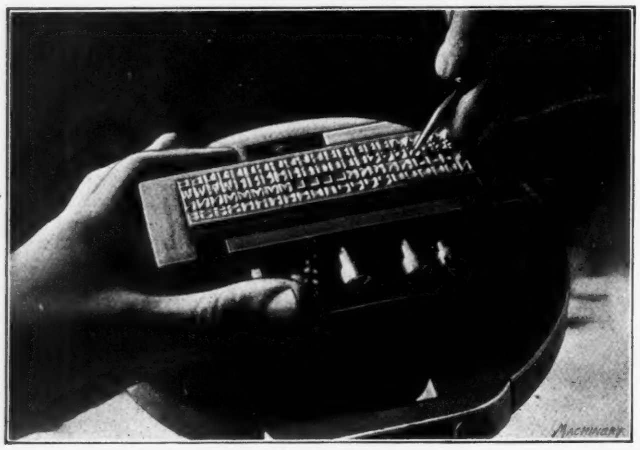


Fig. 8. Using a "Spitz Tool" in Trimming

is employed for cleaning up the insides of the letters, squaring out the round corners left by the cutter and otherwise completing the work. While trimming up the type it is essential that uniform bevels be maintained, that the sides of the letters be free from irregular spots, and that the

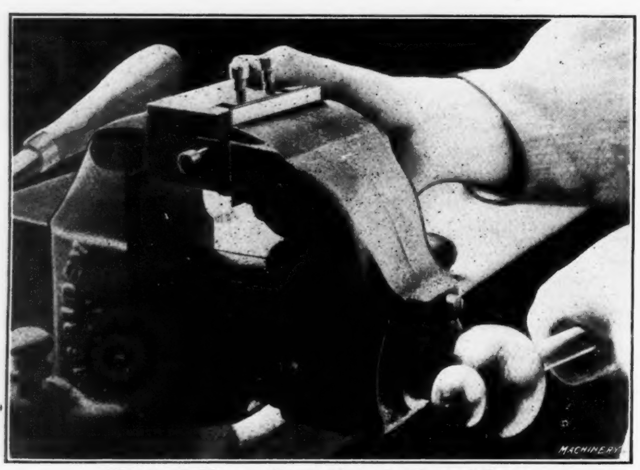


Fig. 9. Taking an Impression of a Font of Type

faces of the type be even; of course, the sizes and proportions of the letters must agree. Obviously, the work requires much skill, and as the metal being worked upon is tool steel the operation is especially trying.

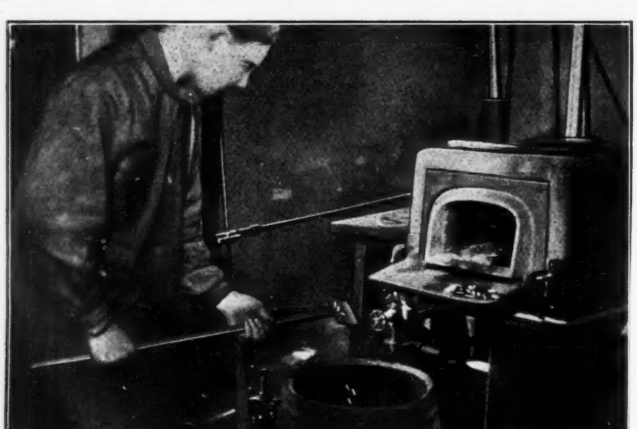


Fig. 10. Hardening the Type

After all the cutting on the work has been completed an impression of the letters in the holder is taken in the manner indicated in Fig. 9. The box of type is held between the vise jaws and a piece of cardboard of the proper quality is placed against the faces of the letters next to the vise jaw and pressure applied to the handle of the vise. This of course causes the letters to be imprinted on the cardboard and by inspecting the resulting impression the workman can tell if the letters are perfectly straight, of the same size, height, etc., more readily than he could by inspecting the type itself. While it is very important that the printing surface of the type blank be maintained, occasionally it becomes necessary to face the end of a piece of type slightly. For this purpose a special facer is used which may be seen at *B* in the illustration Fig. 12. Another case where this facer is utilized is when making up single pieces of type. This facer is a simple affair having a corner into which the blank is pressed, at the same

Fig. 11. Method of using Steel Interchangeable Type

time being held against a stop at the bottom while a file is used on the part of the blank which projects above the hardened top of the facer-jig.

After the engraver has finished the hand work upon the type it is ready to be hardened. This operation is illustrated in Fig. 10 from which it will be seen that the type is heated

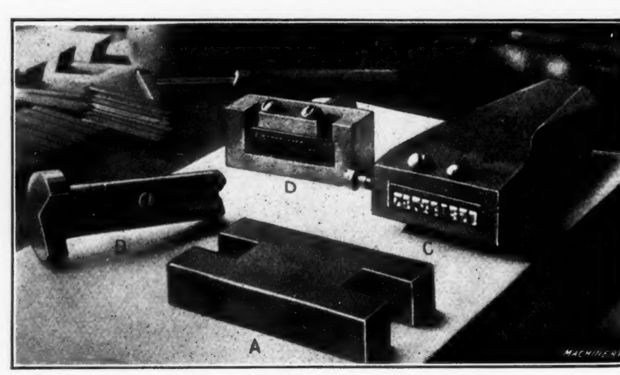


Fig. 12. Type-holders, Facer and Gage

on a thick iron plate, using a muffle furnace. When at a proper heat the hardener slides the pieces from the plate into a brine solution. After the temper is slightly drawn the type is ready for use for stamping or printing.

Using the Type

For stamping metal articles like those shown at *A* and *B*, Fig. 1, the type is used in a foot, screw or power press, according to the character of the work. For printing on cloth or similar material it is used in a printing press in the ordinary way. Fig. 11 illustrates a pressman setting up steel type for printing soft aluminum plates, a foot press being used

for the purpose. The type-holder or chase is similar to the box in which the type is held for cutting, except that it has a hardened steel plate at the bottom and is fitted with a shank so that it may be held in the press. A smaller, but similar holder, is shown at *D*, Fig. 12, and on a larger scale in Fig. 13. Holders are also made for stamping by hand and one of these is shown at *C* in Fig. 12. The only difference between this holder and the one shown at *D* is that it is made from a forging which is provided with an end which may be struck with a hammer. As in the case of the holder shown at *D* this hand stamping holder is fitted with a hardened plate

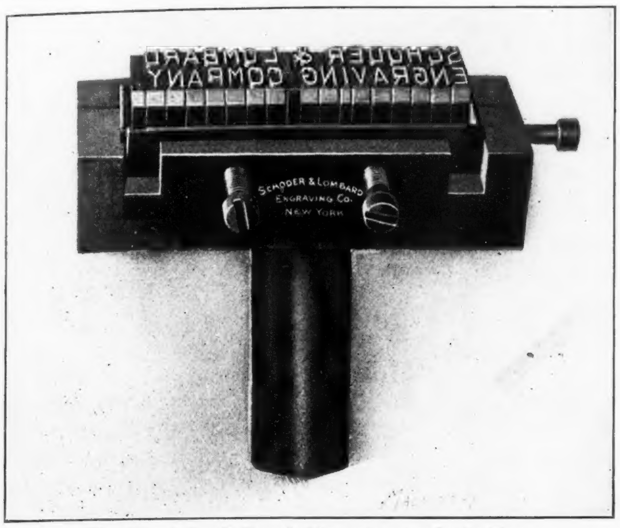


Fig. 13. A Small Type-holder set up ready for Use

at the bottom of the type-holder to prevent the type bodys sinking into the soft steel holder. One of the great advantages of using steel type rather than solid steel stamps is that different words may be set up without going to the expense of having new stamping dies cut. In addition when a piece of type becomes broken it is only necessary to insert a new piece.

Embossing Type

Type for embossing letters in sheet metal is also made along the same lines. Referring to Fig. 1 again, at *C* and *D* may be seen specimens of embossing by means of movable steel type. This type is made the same as stamping type except that the letters are sunk in the blanks. The letters are laid out and routed and then finished by hand in the same manner as raised letters. For forcing the sheet metal into these letters copper "forces" may be made by striking a piece of copper into the previously set up type, or, if the metal to be embossed is too thick to permit of this being done, raised letter type, made to correspond to the sunken

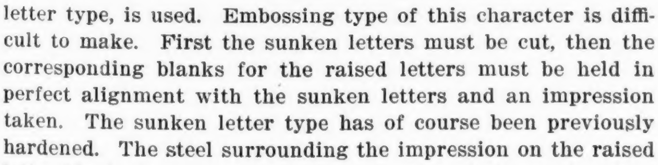


Fig. 14. An Interesting Use of Embossing Type

letter type, is used. Embossing type of this character is difficult to make. First the sunken letters must be cut, then the corresponding blanks for the raised letters must be held in perfect alignment with the sunken letters and an impression taken. The sunken letter type has of course been previously hardened. The steel surrounding the impression on the raised letter blanks is then routed away. By thus fitting each raised letter to its corresponding sunken letter and numbering the pairs, it is insured that good alignment will be secured. With a font of this embossing type metal articles may be embossed with any desired words or lettering.

One Application of Embossing Type

An interesting set of embossing type was recently made by the Schoder & Lombard Engraving Co., for embossing automobile lamp trimmings. The maker of automobile lamps wished to stamp the dealer's name on each lot of lamps ordered, but as the name had to be embossed on the rim of the lamp which surface was also crowned, the dies necessary for each individual shipment were quite expensive, and of course a separate die was required for each name to be embossed. To overcome this die expense the manufacturers made a stamping fixture to fit into a power press. The general design of this fixture is shown in Fig. 14, and as will be seen it consists of two halves; the lower part is fitted to receive the raised letter type and the upper holds the sunken letter type. Suitable clamping plates may be drawn down against the beveled sides of the type. The raised letter type is set up in proper position and the location of the corresponding sunken letter type is adjusted before being finally clamped in position. The work is then treated in the ordinary manner, being laid upon the lower die and the letters embossed between the type. In making the type blanks rings were first turned to the proper dimensions. The blanks were then sawed from these rings being in the form of segments. Two of these pieces are shown in Fig. 1 at *J* and *K*. It will be seen that after once being equipped with these tools this method of embossing was more economical than was the case when the solid dies were used, for it will be readily appreci-

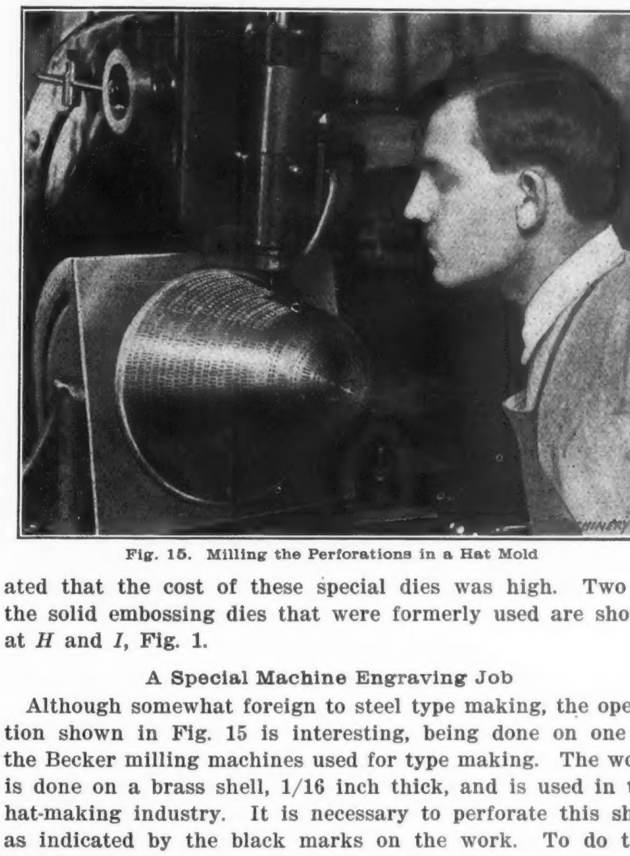


Fig. 15. Milling the Perforations in a Hat Mold

ated that the cost of these special dies was high. Two of the solid embossing dies that were formerly used are shown at *H* and *I*, Fig. 1.

A Special Machine Engraving Job

Although somewhat foreign to steel type making, the operation shown in Fig. 15 is interesting, being done on one of the Becker milling machines used for type making. The work is done on a brass shell, 1/16 inch thick, and is used in the hat-making industry. It is necessary to perforate this shell as indicated by the black marks on the work. To do this the shell is mounted upon an adjustable angle-iron, first being supported upon a wooden arbor. While in this position the cutting is done with one of the tools used for type-making. The work is turned from time to time to present different surfaces of the shell to the cutter, and while the job is tedious it is effectively accomplished on this machine.

* * *

BRITISH INDUSTRIAL CONDITIONS

The British Board of Trade returns for 1912 indicate that the machine tool industry in Great Britain was in a flourishing condition during the past year. The exports of machine tools were valued at about $4,675,000, which was about 20 per cent more than in 1911. The total value of exports of all classes of machinery amounted to over $160,000,000, being an increase of more than $10,000,000 as compared with the preceding year. British shipbuilding also broke all previous records during the past year. There were at the end of the year, in course of building, merchant vessels with an aggregate gross tonnage of about 2,000,000, and battleships with a total tonnage of 500,000.

SKILTON DIGS SOME CLAMS—A VACATION STORY

BY A. P. PRESS

Skilton had a bug—not the good old-fashioned kind that we always have with us, and which the poem says, "has no wings, but arrives at its destination just the same." Skilton's bug was an entirely different kind, and its name was "Scientific Management" or, as he expressed it: "The Conservation of Energy."

It is true Skilton had good reasons for having this insect, for the nature of his work forced it upon his mind continually, but Skilton's theory was that the same practice could be applied to the everyday walks of life—in the kitchen, down cellar after coal, and in fact, there was no limit to its application.

He went through the square in this way and every clam * * * was deposited in the bucket with one motion of the arm

So when the middle of August came around, and he and Mrs. Skilton and the three little Skiltons went on their annual vacation, he proposed to put his theory into practice. They went, as all good city people do, to the seashore, only it was to the real seashore where you get your clams with a digger and bucket, and do not cross the palm of the native with silver unless you are forced to when the fish do not bite.

The first morning after breakfast Skilton proceeded to demonstrate his theory down on the clam flats. His idea was that if he could get next to a native, he could work out his idea without saying anything and prove it to be just what he had said it was.

Equipped with a round bucket, because mathematics had proved to him that a round receptacle held more in proportion to its weight than any other shape, and a new clam digger, he started for the flats. Half way down to the beach he fell in with a barefooted boy, who seemed to consist chiefly of a long pair of legs, encased in overalls much too large, and a shock head of hair that would have made John D. (and some of the rest of us) green with envy.

"Going clamming, sonny?"

"Well I guess so."

"Guess I'll drop in with you."

And "drop in" Skilton did, and the two meandered down to the clam flats. Getting down to the beach Skilton proceeded to put theory into practice. He marked out a square of about ten by twenty feet, and started down in the lowest corner (it is easiest to dig up hill, you know). He took just the same amount of earth with the digger each time, and reduced his motion so as to use the least possible amount of muscular exertion to each one, and when he came to a stone, removed it with the same care and put it one side.

He went through the square in this way and every clam, when he found it, was deposited in the bucket with one motion of the arm. He got through his track and counted his clams and he had thirty-seven, and then started over where "Legs" was working. The boy had his basket nearly full, but a look at the beach disgusted Skilton. He had dug here, there, and everywhere, and when he got to a rock he just let it be. (He knew there were no clams under it.) And when they did not come out thick enough he had gone on to a new spot. A shiftless way, thought Skilton, but it certainly brought results.

"Kind of dirty, aren't they, sonny? Think we had better wash them?"

So out in the clear water they both waded. The boy did not seem to mind it, but it did not feel good to Skilton to get his feet wet. Skilton poured out his clams, dipped in his pail, rinsed it out two or three times, picked up his clams again, and pronounced them clean. The boy dropped his basket into the water, gave it a shake and a swirl, and picked it up with just clams, and no dirt left, and they started for home. Going back of the last bunch of sea-weed, Skilton says:

"Say, what will you take for the clams?"

"A quarter."

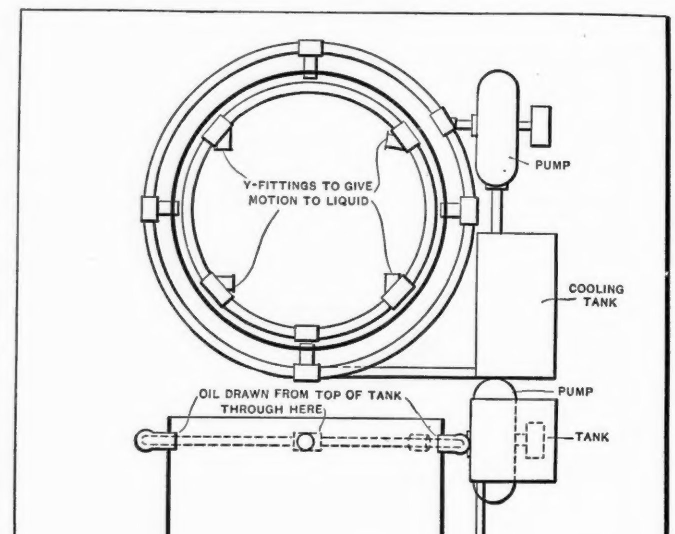
There was an exchange of commodities and Skilton put his thirty-seven clams on top of the others and went home.

Mrs. S. was pleased, and they had the clams for dinner. But scientific management, as applied to digging clams, was not served at the table the rest of the vacation.

* * *

AN OIL QUENCHING TANK AND COOLING SYSTEM

The accompanying illustration, taken from *American Vanadium Facts*, shows a cylindrical quenching tank and cooling system used by the American Locomotive Co. for quenching engine parts. The chief feature of interest of this tank is the means provided for keeping the oil at a uniform temperature. The hot oil is drawn off from several points about two feet below the surface and is pumped through a series of coils, submerged in running water, then back through the tank at points close to the bottom. The tank is set within another sheet-iron tank sunk in the earth, and is so placed that its top is a few inches above the floor level. The circulating water is pumped by a No. 2 Gould rotary pump through 250 feet of two-inch pipe. The cooling tank as shown in the illustration is not drawn to scale. With this equipment, anywhere from thirty-five to forty tons of forgings can be quenched in twenty-four hours.



Oil Quenching Tank and Cooling System used by the American Locomotive Co.

* * *

The approximate output of automobiles in the United States durnig 1912, was 275,000; the estimates given vary from 250,000 to 285,000. Judging by the reports from several automobile companies this output should be considerably increased in 1913.

THE HUMAN FACTORS IN ENGINEERING PRACTICE*

ORGANIZATION METHODS, PRINCIPLES OF MANAGEMENT, ETC.

Organization is recognized as an economic necessity to the effective control of cooperative human effort. Organization deals with men rather than with things, and industrial organization includes the elements of production, transportation and commerce. The business problem in any undertaking is to limit expenditures in such a way that a satisfactory margin of profit may be preserved. This requires the most effective team work, both on the part of the individuals who constitute the officers of the enterprise and those who form the rank and file of the industrial team. Organization is the art of uniting and directing the working forces in order that they may produce the most satisfactory result. It is the duty of the officials not only to operate a plant economically, but also to anticipate business fluctuations, to measure the productive value and desirability of extension, to check mere increase of size, and to make reasonable provision for a contracting expense of organization during periods of depression.

In a certain sense, the individual must organize his efforts if he is to be in any way certain of success. To succeed, he must first have a clear understanding of what he desires to do; second, he must determine the method he will use, and third, he must act constantly and persistently in the prosecution of the method which is to secure the desired result. Frequent changes in the method of precedure—even where such changes are for a better plan—will not lead to success. What is true of individual effort applies even more to the action and reaction of individuals in their authority over one another, and with the substitution of an organization for an individual, new difficulties arise. The individual suffers only from his own lack of ability, but where more than one person are concerned, there are added dangers of misunderstanding, of clashing opinions, of a diversity of interest, and of insubordination and intrigue. Organization cannot entirely eliminate these difficulties, but it can reduce them in direct proportion to its degree of efficiency.

The question of centralizing authority is one which requires careful consideration, and there are two extremes between which the most desirable results lie. If the officials are entirely unrestricted in their actions the interest of the industrial organization is often lost sight of and harmony of action becomes impossible. If, on the other hand, all power of initiative is placed in the hands of a central executive, he will have such an amount of detail to look after that great delay is unavoidable, and it will be impossible for him to study varying conditions which arise, with a result that general rules are applied in cases requiring special attention. Under such management the emergencies which arise in every industrial undertaking are handled in a feeble and inefficient fashion, and a disastrous breakdown of the organization is almost certain to occur.

Organization Methods

There are two important methods of handling the organization problem or a large undertaking having a number of plants—the divisional and the departmental methods. Under the divisional method, single plants and enterprises are given a more or less complete organization under one chief executive officer. The departmental method, on the contrary, provides an officer in charge of all similar departments regardless of where they are situated, with subordinate officers who report to him from each place. In actual practice the divisional plan of organization is not completely carried out, as motives of economy and varying local conditions do not permit of the sharp separation of work at different places. With the departmental plan of organization, this difficulty disappears, but others take its place. As a rule, no one department is sufficient unto itself, as its work requires the cooperation of other departments. Under this method of organization it is more difficult to eliminate any lack of harmony, because reference has to be made to a common superior officer who is not so closely in touch with local conditions. The divisional method of organization tends to develop more broadly trained men for the higher positions than is possible in the departmental plan. Such men have "gone through the mill" and have been brought closely in touch with the workings of departments other than those in which they have been employed, the result being that they have a broader outlook upon industrial conditions.

The Necessity for a Leader

The prime object of organization is to bring brainy men together for work and action, and a wise organization seeks and engages men of ambition. It is realized that success demands an aggregation of men of strong individuality, who are free to contribute their quota of wisdom, but who make their individual preferences secondary to the general policy of the organization. In order that work may be well done, and to have its action strong and forcible, the organization must move as a harmonious unit, and mere scheming can never lead to ultimate success. Herein lies the task of the leader—the organization of men, and the systematizing of things. Both of these factors are important and it is a fortunate coincidence when one individual combines capability in both directions. The principles followed in a good organization may be outlined as follows: 1. An executive with power to decide on all points where action must be taken is provided. 2. The responsibility of each position is fully and carefully outlined, in order that misunderstanding may be avoided. 3. The duties of the different organization positions are made to conform satisfactorily to the ability of those chosen to fill them. 4. No person is made subordinate to two or more superior officers, if it is possible to avoid such a course. 5. The power to discipline men in any department rests in the hands of the official who is held responsible for results. 6. The duties of the members of the organization are distributed so that a fair comparison of results is possible. 7. Wherever possible no positions on the staff are created which offer no further opportunity of promotion for an ambitious man.

The Place for the Engineer

The last twenty years of industrial development have been characterized by a steady advance in the part which the engineer has played in this field. In all branches of industry men of purely clerical and commercial experience have been replaced to a considerable extent by engineers. The result has been that such men, aided by their sound scientific and practical mechanical knowledge, have directed their energies toward a careful analysis and accurate solution of industrial problems, and where these men have also developed executive ability, the combination of self reliance, leadership and trained judgment has placed many engineers high in industrial organizations. At present many of our technical schools are training men to fill just such positions, and they are designated "engineers" for lack of a more adequate term.

Many an engineer has failed in organization work because of his limited knowledge of human nature and his inability to deal intelligently with his fellow men. Such men are content to be individual designers of distinction, or consulting engineers on purely mechanical matters, and in such work a lack of broad knowledge of human nature doesn't hurt them professionally. But this shortcoming certainly limits their opportunity of advancement to the highest positions in industrial work. Of all requirements of the engineer who is seeking a place in an industrial organization, there is none of such vital importance as executive ability—the art of getting things done. This should not in any way be confused with personal ability to do things. Executive ability is closely allied to character and may be developed to a certain extent, although it is originally a natural gift. There are many men who can invent, men who can clothe a scheme in engineering dress, who can carry financial and commercial preliminaries to a successful close, but who are unable to get the scheme itself carried out to a satisfactory conclusion. Many young engineers find themselves in the unfortunate position of knowing how to do a thing without being able to get it done. When the project is finished on paper some men have lost all interest

*Abstract of lectures delivered to senior students of the Stevens Institute of Technology, Hoboken, N. J., by John Calder, of the Cadillac Motor Car Co., Detroit, Mich.

in it, so that the idea for them is merely a creation of the mind without any real significance. Such men may be good calculators or excellent estimators, but they are not engineers. The engineer is the man whose task interests him right up to the final step which brings it to completion.

Cooperation

The ability of the young engineer must be supplemented by the faculty of getting along harmoniously with his fellow executives. It may sometimes appear advantageous for a man to hold aloof from his fellow workers, and in rare cases this may actually be true, but failure to cooperate heartily with his fellows is a fatal error to the ultimate advancement of an engineer. No man ever failed in life through doing too much for others. Team work is the essence of modern progress and the higher industrial executives are now more on the alert to perceive and to reward it than over before.

Management of the Modern Plant

The modern element in the art of management is the adoption of what has long been known as the scientific method. Its industrial applications cover a wide field and a writer defines it as follows: "The critical observation, accurate description, analysis and classification of all industrial and business phenomena of a recurring nature, including all forms of cooperative human effort, and the application of the resulting records to secure the most economical and efficient production and regulation of future phenomena." In other words, all repeated experience with men and methods is criticised, carefully verified and recorded, and by the three methods of logical inference, *i. e.*, analogy, induction and deduction, is made to yield new results ready for application wherever they appear to be justified. Utility is the only justification for expenditures on scientific methods of management and there are quite a few of its results which cannot be economically applied to industrial conditions. The ability to avoid the application of new methods in cases where they are not suitable is lacked by men who have not had wide shop experience. Previous to the application of scientific management, observed differences in the efficiency of plant operation were largely due to the varying ability of managers.

In 1897—sixteen years ago—there were signs of an increasing number of technically trained men in the practical control of industrial plants, as distinct from laboratory work. This marked the unformulated and unnoticed beginning of the new element in management, and quite a few of the shop executives had been mentally prepared for advancement before the new era had been recognized. An analysis of the published matter on this subject, made by the writer at the time when the new movement began, shows that the directors of the metal working trades in particular had followed the path of least resistance and applied new methods largely from the accounting side of the business. This resulted in more reliable inventories and cost systems, and brought out the causes of inefficiency to a limited degree, so that it was possible to inaugurate reforms. But comparative cost systems could not establish standards of shop performance and methods of obtaining them; and although the accountant was first in the field, he lacked ability to deal with the work and was later supplemented by the experienced engineer, with an all-round training and a new point of view regarding the problem of management.

The Regulative Principles of Management

This new element in the art of management has been defined as the use of the scientific method in industry. The general problem of industrial establishments is economy, and may be defined as follows: "To furnish daily the prescribed quality and quantity of product in all its varieties by most efficient shop and labor arrangements, and with a minimum of cash capital locked up in the process." A productive organization and system of plant management which will accomplish this and continue to do so with harmony and to the satisfaction of both the employer and the employe attains the desired end. In problems of management the most satisfactory results are derived from observing the three following classes of industrial conditions. 1. The economic results of different arrangements and forms of materials and all operations upon them. It is the object of the scientific method to make the most of past experience to see that efficient methods are used. 2. The economic results for varying executive methods for effectively directing human effort as a whole in the use of the data obtained from observation of the operating conditions. 3. The economic results of steps taken to raise the industrial efficiency of the individual worker in every grade of service. This covers the entire problem of labor, reward and intensified activity. By carefully considering these conditions and the data derived in this way, it is possible in many cases to formulate a scientific basis for economically handling business and industrial conditions which recur periodically.

Review of the Past Ten Years

Ten years ago the recognized divisions in management were being modified more or less consciously under the influence of the scientific method. In varying combinations these divisions have appeared in specific "systems" during the past ten years combined with some special method of labor reward or incentive; but essentially they are all industrial practices of long standing. It was at this stage of the movement that a number of accountants, followed by engineers, specialized in the promotion of industrial efficiency and from time to time published their "systems" of management. Foremost and most important of these "systems" was that now termed "scientific management," which F. W. Taylor advanced in 1903 as an outcome of his researches into the art of cutting metals. In many ways his experiments differed in no important essential from the previous investigations for determining other mechanical and physical constants, but the large number of variables which were involved rendered his labors exceptionally difficult and tedious and had hitherto deterred engineers from attempting a solution. They led in particular to a scientific re-arrangement of men, materials and operating methods, upon which was based the generalization that the laborious and expensive task of framing a true science for every element in industrial problems was not only possible, but an absolute necessity as a preliminary to securing higher efficiency. Mr. Taylor went even further and claimed that the main elements of his experimental efforts were identical with those required for the solution of a vast number of other and quite different shop and business problems.

Many factory executives, who are convinced that the application of the scientific method is the inevitable and natural course in the solution of industry and of business, are by no means agreed that Mr. Taylor's solution, or indeed any of the formulated systems which have followed it, are equal to the requirements of industry as a whole. They believe that before the scientific method can be made capable of general application, it will be necessary for it to find application in a large variety of industrial problems from which experience may be secured.

Some Reflections on Ways and Means

It can scarcely be said that the Taylor system has made any impression outside of the machine shops and it has gone only a short way in that branch of industry. But this does not reflect discredit upon the scientific method, which must always face different conditions in industry from those met with in science, and constantly yield to and be measured by consideration of ultimate utility, of available capital, and of current prices and profit. It is futile to expect any future in industry for a scheme of management which uses science to the smaller benefit of the investor, although such a course may be followed by the zealous systematizer. It is seldom desirable to commit a concern to an elaborate program of re-organization which involves years of transition experience and expense. It is necessary for men believing in scientific management, but more conservative and experienced than the ardent systematizer, to hold the reins of business and "guide the team." An undesirable condition has arisen through the efforts of practitioners in methods of management to promote their business as rapidly as possible. With this end in view, figures have been published showing remarkable increases of efficiency that have been secured in certain cases, but without explaining what the conditions were before the new system was made effective. The latter is necessarily a most important point in enabling an estimate to b formed of the improve-

ment that was effected through the substitution of a new system of management. There is perhaps no more useful service which the engineer can render in industrial work than to secure and circulate information which will convince factory owners and executives that the scientific method opens up no royal road to success, but that it simply affords a most practical and reasonable course toward the efficient control of industry.

* * *

MILLING KEYWAYS ON A PLANER

BY H. A. STENGER*

The illustrations show a Cincinnati planer which has been equipped with an attachment for milling keyways in shafting. In Fig. 1, a shaft is shown clamped to the table in three V-blocks ready for cutting two keyways $1\frac{1}{4}$-inch wide by $\frac{1}{2}$-inch deep and about 20 inches in length; these keyways are located 18 inches from each end of the shaft, and they can be cut at one setting of the work and one cut. In using this attachment, the cross-rail of the planer is lowered so that it just clears the clamps used to hold the work. The

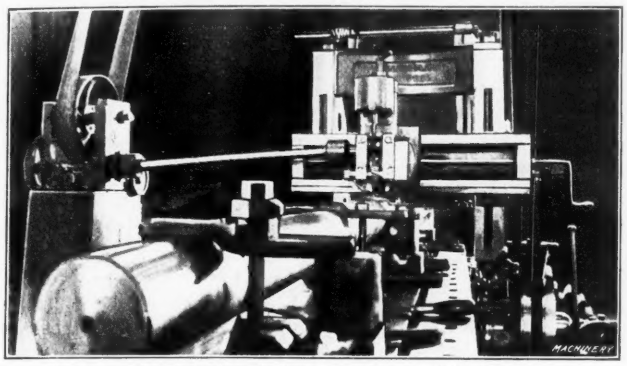


Fig. 1. Planer equipped with Fixture for milling Keyways

milling cutter is held on a shaft running in a bracket bolted to the planer head, the toolpost being removed from the head and the bracket bolted in its place. The bracket used for this purpose was made from a broken crankshaft.

Fig. 3 shows a near view of the driving mechanism which is set up at the left-hand side of the planer for driving the milling cutter and feeding the work. This mechanism is belted to a countershaft and is made from an old locomotive driving-box. It is set up on a stand about twelve inches higher than the table of the planer and at a distance of 3 feet from the bed of the machine. It will be seen that this drive has two shafts, the upper one being belted to the counter-

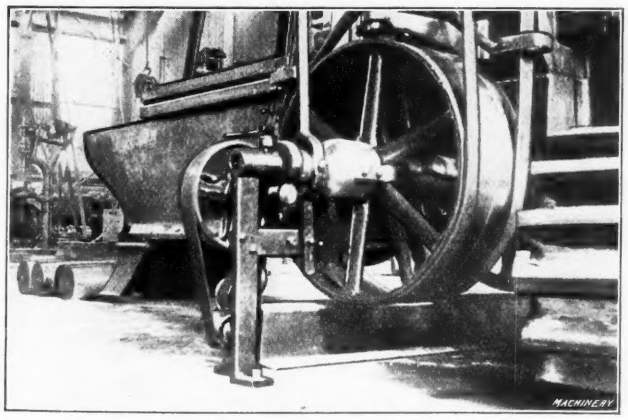


Fig. 2. Belt-driven Reduction Gear for feeding Table

shaft and connected to the lower shaft by gearing. The lower of these two shafts is connected to the upper shaft of the milling attachment by means of a universal joint, and drives the milling cutter mounted on the lower shaft of the attachment through gearing.

A small shaft will be seen extending out at the back of the driving mechanism in Fig. 3. The $1\frac{1}{2}$-inch belt running over the pulley on this shaft passes under the bed of the planer and drives the loose pulley through the worm-wheel connected with its hub by a clutch. The belt on this pulley runs up to the planer countershaft and drives the tight pulley by means of the second belt. For milling keyways with this attachment a very slow table travel is necessary and the drive is arranged to give the required speed reduction. When it is desired to reverse the direction of table travel, the clutch is knocked out and the belt shifted; the table can then be backed away by pulling the belt by hand.

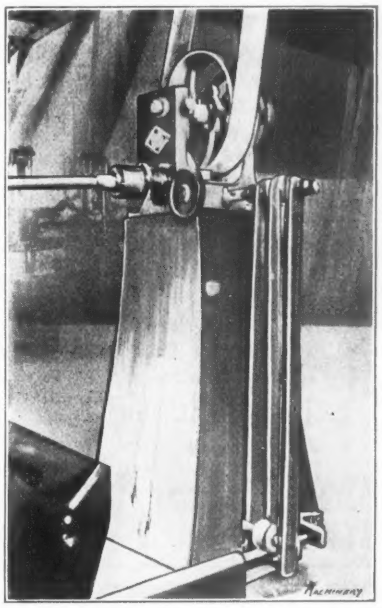


Fig. 3. Device that was used for driving Cutter and Power Feed

The interesting feature of this attachment is that it is made almost entirely from material taken from the scrap pile. It has already been mentioned that the bracket for the milling attachment was made from a broken crankshaft and the drive from an old locomotive driving-box. Most of the gears that were required on the milling attachment and drive were available among our supply of lathe gears, so that it was only necessary to cut a few small pinions for use in this connection.

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* * *

CHROME-VANADIUM STEEL WHEELS

The results of comparative tests made on chrome-vanadium and carbon steel tender wheels on the Vandalia R. R. were recently made public. The tests show that chrome-vanadium wheels average $2\frac{1}{2}$ times as much mileage per unit diameter of wear and turning as do carbon steel wheels in the same service. The average load per wheel was 17,875 pounds. The average mileage per 1/16 inch diameter wear was 12,857 miles for chrome-vanadium wheels and 9249 miles for carbon steel wheels; the average mileage per 1/16 inch turning was 6894 miles for chrome-vanadium steels and 3544 for carbon steel wheels.

The valuable characteristics of chrome-vanadium steel wheels are uniformity of material, absence of shelled or burned spots on tread, less flange wear, less material wasted in turning wheels in the lathe and consequent greater mileage per inch diameter of wear plus turning. The physical properties of the chrome-vanadium wheels are: Elastic limit, 98,000 pounds per square inch; ultimate strength, 140,000 pounds per square inch; elongation in 2 inches, 7 per cent. It is the general opinion of wheel experts that the capacity of a wheel or tire to resist wear and shelling depends on the elastic limit of the material. When shelling or kindred troubles occur they can be greatly reduced or practically eliminated by using steel having a sufficiently high elastic limit and other physical properties in proper ratio. The high elastic limit of chrome-vanadium steel used for tires is indicated in the foregoing summary of ten tests.

The wheels are subjected to special heat-treatment, developed as the result of research. The heat-treatment consists of heating the wheel to a temperature of 800 to 875 C, and then spinning it for a predetermined time in a trough of water with the tread immersed to a point slightly below the wear line limit of the wheel. The wheel is then taken out and cooled in air, after which it is reheated to 600 to 700 C, for annealing. By this method of heat-treatment the plate and hub of the wheel are not affected by the immersion of the rim in water, except that they undergo an annealing operation as a result of the hardening of the rim.

* * *

The steam engine has been threatened with extinction a number of times, but apparently it is not frightened. It seems to hold its own very well.

STEAM POWER PLANT PIPING DETAILS—3

MAIN STEAM HEADER DETAILS

BY WILLIAM F. FISCHER*

The general arrangement of the piping at the main steam header is governed, to a large extent, by the arrangement of the boilers and engine units, and by other local conditions such as the general design of the building, structural details, etc. In this article, the writer will not attempt to show the complete design of a main steam header with its boiler and engine connections, methods of supporting mains, etc., but will describe the details at the header for one or more units and several methods of arranging the piping to accomplish certain results, leaving the reader to adopt what he feels is the best method for any specific case.

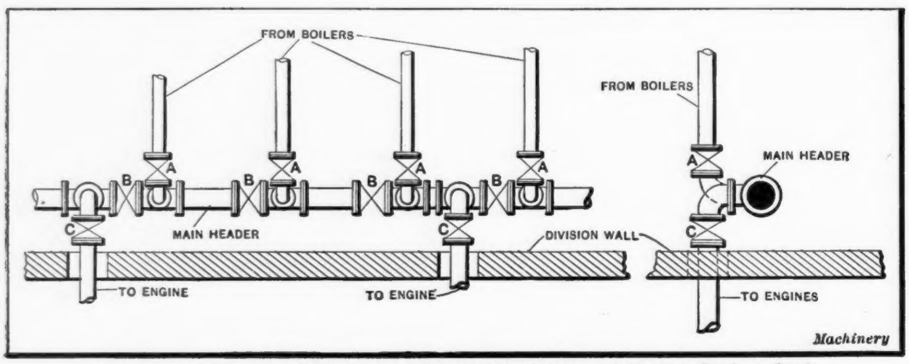

Fig. 13. Arrangement of Piping for a Plant with Boilers and Engines Back to Back in Separate Buildings

In Fig. 13 is shown a method of arranging the main steam piping in a power plant in which the boilers and engines are placed back to back in separate buildings with a division wall between them. As stated in the first installment, the main steam header, and as much of the steam piping as possible, should be placed in the boiler room so that a blow-out will not damage the electrical apparatus. In this case, the main header is run along the division wall at the rear of the boilers. The boiler branches make connection to the top of the header through the gate valves *A* and elbows, as shown.

The main header is protected by the gate valves *B* placed between each pipe connection to and from the header. The steam for the engines or turbines is taken from the top of the header through elbows and gate valves *C*. If so desired, the gate valve and elbow at each connection to or from the header may be replaced by an angle valve. The object of the valves *B* is to make it possible to isolate any section of the main header in case of accident, so that the plant will not have to be shut down while repairs are being made. This subject was discussed at more length in the first installment. The end elevation of the piping shows the connections to and from the header. One objection to this system is the large number of valves *B* that are required, and as these valves are quite costly, they are frequently placed between every third or fourth connection to the header instead of between each connection. In some cases these valves are omitted entirely, their use in any specific case depending upon the designer's judgment as to whether they are required or not. In central power stations where a blow-out in the steam main may seriously cripple the lighting and power system of an entire town or city, it is advisable not to take any chances of a "shut down." In any case, the designer should study the conditions carefully before deciding what arrangement of piping and valves to adopt for any particular case. Another objection to this system is the large number of flanged joints in the main header that must be made up with gaskets and repacked when leakage occurs.

In Fig. 14 is shown another method of arranging the piping at the main header. In this case, the branch pipes to and from the header make connection to the top of manifold fitting *E*, as shown, and each branch pipe is equipped with a straightway gate valve and elbow where it makes connection to the header. As previously mentioned, this gate valve and elbow may be replaced by an angle globe valve, if so desired, but the writer favors the construction shown in Fig. 14. A gate valve is placed at each end of the manifold fitting *E*,

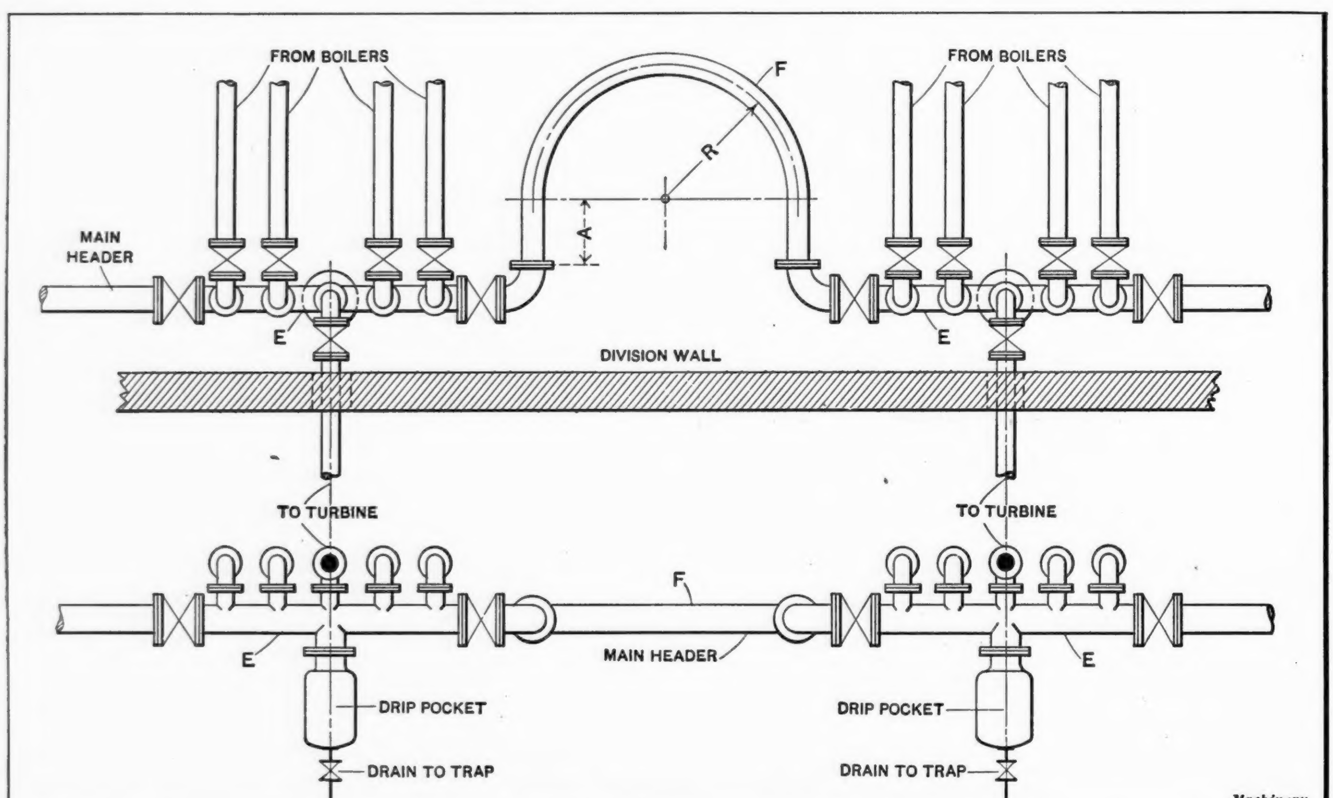

Fig. 14. The "Unit System" of Piping; in this Case Four Boilers and One Turbine constitute the Unit

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thus dividing the main steam header into independent units or sections, making it possible to cut any section of the header out of service as desired. This piping is arranged on what is known as the "unit system." In this particular case, four boilers and one steam turbine is considered a complete unit, since four boiler units are required for the operation of each turbine unit. Therefore, if the valves at each end of the manifold fittings are kept closed, the piping system will be operated on what is known as the "unit system of operation." The steel manifold fitting is cast in one piece, thus doing away with a considerable number of flanged pipe joints in the main header, and a drip pocket should be placed beneath each manifold fitting, as shown. This drip pocket will keep the piping system free from water, thus preventing water-hammer and similar troubles in the piping system and in the engine cylinders. The expansion bend F provides for expansion and contraction in the main header and prevents the strain from falling on the pipe joints. This bend should be made from a piece of "full weight" steel pipe, heated in a furnace and curved or bent to the required dimensions. In order that this bend be made as flexible as possible, the radius R should be as large as existing conditions permit, and never less than six times the diameter of the pipe from which the bend is made. For the same reason, the straight ends of the bend A should be as long as possible, and not less than $1\frac{1}{2}$ times the diameter of the pipe.

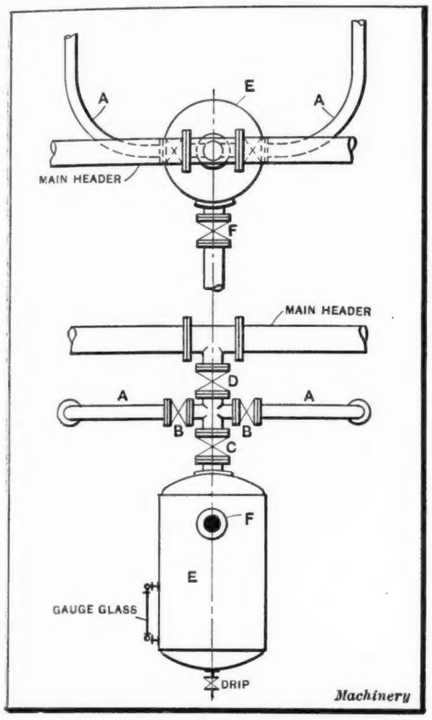

Fig. 15. Another System of Piping for Unit Operation; the Main Header acts as Tie-line between Units

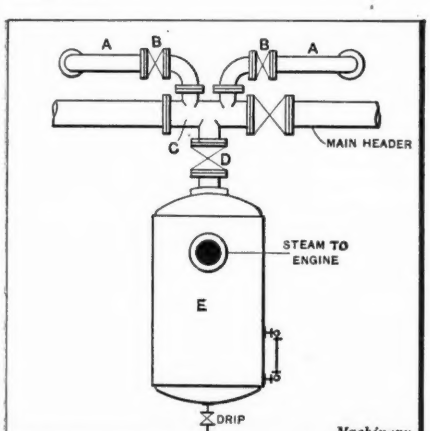

Fig. 16. Arrangement of Piping where Separator drains Condensation from Main Header

Another method of arranging the piping at the main header for independent unit operation is shown in Fig. 15, the main header acting merely as a tie-line between the several different units. In this case each engine or turbine has its own complement of boilers which discharge into the boiler headers A. These boiler headers or branch pipes are brought together at the main header through the valves B, the steam from the boilers entering the separator E through the valve C and passing on to the engine or turbine through valve F. Ordinarily, the valve D is kept closed, thus isolating the main header. In this way, each unit may be operated independently of the others and at its most efficient and economical steam pressure. Thus, the boilers for one unit may be operated at 125 pounds gage pressure, another unit at 150 pounds gage pressure, and another at 180 pounds gage pressure, as desired. (By gage pressure is meant the steam pressure in pounds per square inch as registered on the boiler steam gages.) If it is desired to shut down any one engine unit, and use the steam from the boilers temporarily to supply an engine or turbine of any other unit, valve F may be closed, valves B, C and D opened, and the steam passed through the main header to the desired engine unit. By leaving valve C open, any water carried over from the boilers will flow to the

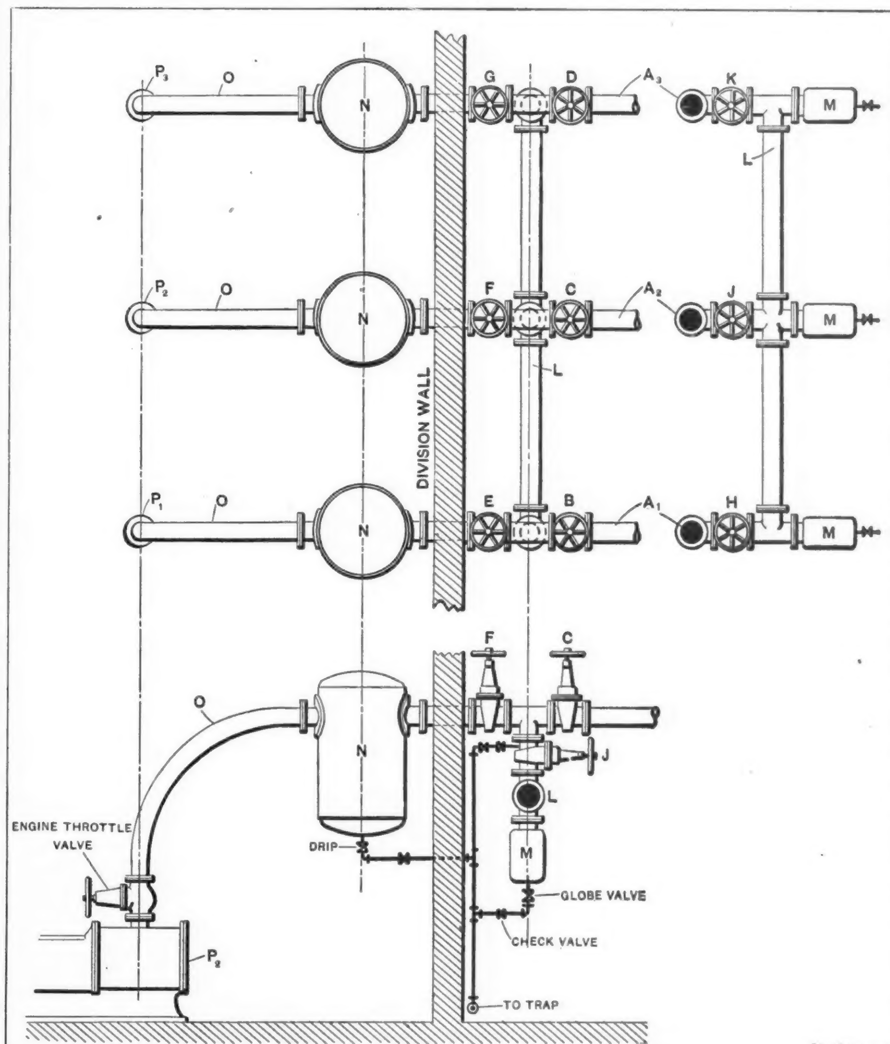

Fig. 17. Arrangement of Piping for Unit Operation, showing Method of diverting Steam from One Boiler to Different Engine Units

separator and not be carried over into the main header with the steam. In this particular case the steam separator E is quite large, being of the receiver type, and for this reason it was placed in the boiler room as shown, rather than in the engine room.

In Fig. 16 is shown another method of arranging the piping at the main header. In this case the steam from the boilers enters at the top of the main header through bends A and valves B as shown. A special manifold tee C is used, which takes the place of three standard tees and reduces the number of joints in the header. The separator E keeps the header drained free from water during operation through the valve D.

Still another method of arranging the steam piping system for independently operated boiler and engine units is shown in Fig. 17, where the boiler headers A_1, A_2 and A_3 supply steam

to the engine units P_1, P_2 and P_3. In this case each engine unit has its own complement of boilers, and the main header *L* acts merely as a tie-line between the different units; this header is protected by drip legs *M* to keep the header and piping drained free from water during operation. These drip legs and also the steam separators *N* (shown on the engine room side) should be drained to a steam trap, pump and receiver, or other suitable device for returning the water of condensation to the boilers. When each engine unit is to be operated independently of the others, valves *H*, *J* and *K* should be closed, thus isolating the main header *L* entirely. If it is desired to pass the steam from any one set of boilers to any one engine unit, this may be accomplished by opening and closing the proper valves at the main header. As an example of the flexibility of this system, assume that engine units Nos. 1 and 2 are in operation, each engine being supplied by its own complement of boilers, and that boiler and engine units No. 3 are out of service at the time. In this case valves *E*, *B*, *F* and *C*, are open and valves *H* and *J* closed, permitting the steam from boiler headers A_1 and A_2 to flow direct to engine units Nos. 1 and 2, respectively. Now, assume that it becomes necessary to shut down engine No. 1 and put engine No. 3 in service in its place, using steam from boiler header A_1 without interfering in any way with the operation of engine and boiler units No. 2. In this case, valve *J* being already closed, it will be necessary to close valves *E* and *D*, and open valves *B*, *H*, *K* and *G*; engine unit No. 3 is then ready for service, using steam from boiler header A_1 in place of A_3. Valves *H*, *J* and *K* should be drained above the seat, as shown in the end elevation, as they form a water pocket in the line when closed. The steam separators are shown in the engine room, in this case, but they may be placed on the other side of the division wall (in the boiler room), if so desired. A vertical type of separator may also be used, having a top inlet and bottom outlet, in which case it could be placed directly above the engine throttle valve. Bends *O* should be curved to a long radius in order to provide for expansion and contraction strains in the piping system.

Steam Connections between Main Steam Header and Prime Movers

In the second installment of this article, in Figs. 6 and 7, examples of incorrect and correct piping connection between header and engines were illustrated and discussed. In Fig. 17 another method of connecting the engines with the main steam header is shown. In this case the steam separators are placed in the engine room close to the division wall, and long radius bends *O*, leading from the separators to the engine throttle valves, provide for expansion and contraction of the piping on the boiler room side.

Fig. 18 shows two different methods of connecting the engines with the main steam header in a power plant in which the engine room floor line and the boiler room floor line are on different levels. In this case the engine room basement floor line is on a level with the boiler room and the engine room floor line is at a higher level. Referring to the piping shown in full lines, the steam from the boilers enters the main steam header through bends *A*. The main header is placed at the rear of the boilers, as shown at *B*, and may be supported by brackets bolted to the division wall, or hung from the roof beams overhead (not shown). From the main steam header, the steam is taken off to the engines through valve *C*, bend *E*, steam separator *F*, riser *G*, bend *H* and engine throttle valve *J*. In a steam line connecting the engine with the main header, a valve should always be placed close to the header, as shown, so that the line may be shut down when necessary to pack joints, etc. The throttle valve at the engine is usually supplied by the engine manufacturer. The steam separator *F* is of the receiver type, and as it is placed in the engine room basement, at the lowest point of the piping system, any water of condensation forming in the main header and branch pipes will drain into the separator, thus keeping the main steam header free from water, and, at the same time, preventing water from being carried over to the engines. In this case the steam connection is taken from the bottom of the header, and as valve *C* forms a water pocket in the line (when closed) it should be tapped above the seat and a drip connection provided as shown at *D*. In a case of this kind the designer may find it more convenient to place the main steam header a few feet above the boiler room floor line, as shown by the dotted lines at *K*, in which case the valves and piping are more accessible than when placed overhead as shown at *B*. With the steam header located as shown at *K*, the steam line to the engine may be taken from the top of the header through angle valve *L* and the straight pipe *R*, which makes

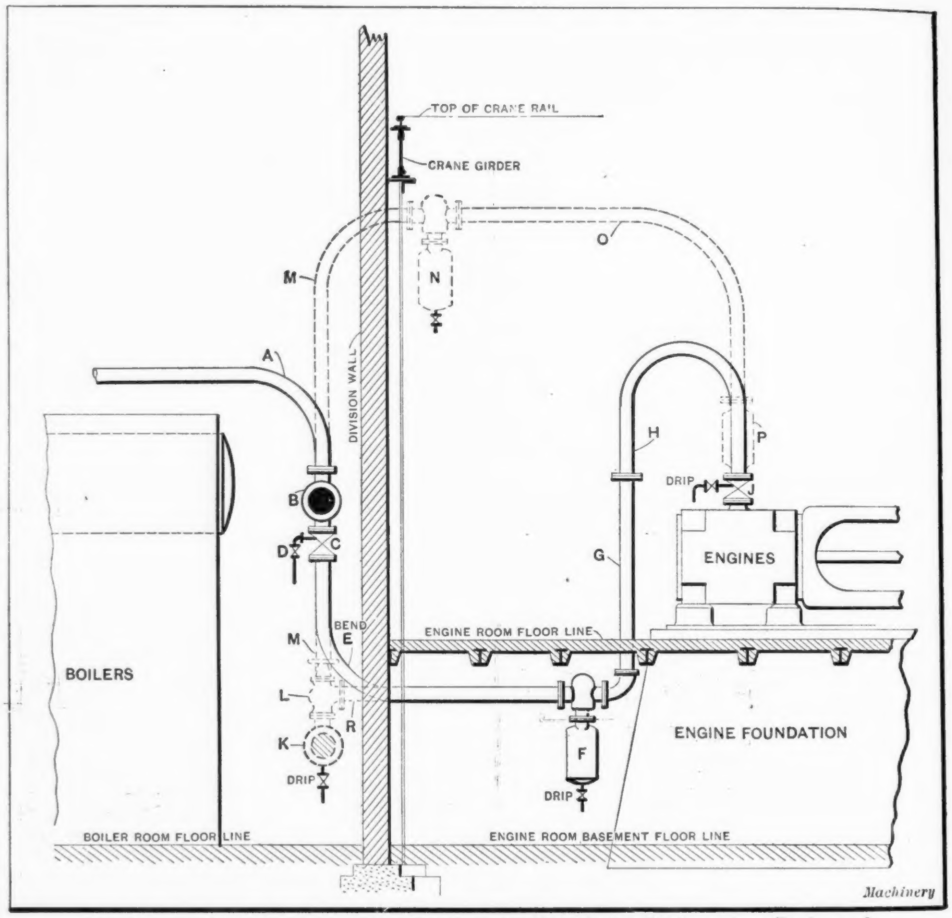

Fig. 18. Two Methods of connecting Engines with Main Steam Header, where Engine and Boiler Room Floors are on Different Levels

connection with separator *F*, the piping beyond this point remaining the same as in the previous case. The main steam header should have drip connections provided at the bottom, as shown, and the drips should be piped to a steam trap or other device for returning the condensation to the boilers.

The dotted lines in the engine room show the steam piping to the engines carried overhead. The steam from the main header is taken off through bend *M*, separator *N* or *P*, bend *O*, and throttle valve *J*. If the horizontal type of separator is used, as shown at *N*, the vertical leg of bend *O* above the throttle valve will gradually fill up with water when the valve *J* is closed. To prevent this occurring, the throttle valve *J* should be tapped above the seat and a drip connection provided. The vertical leg of bend *H* (above the throttle valve) is also likely to fill up with water, due to condensation in that section of the piping when valve *J* is closed.

With the overhead system of piping (shown dotted) a vertical type of separator *P*, placed directly above the engine throttle valve, is preferable to the horizontal type indicated at *N*.

THE REMINGTON "NEGATIVE ANGLE" SIGHTING SYSTEM

A NEW METHOD OF SIGHTING MILITARY AND SPORTING RIFLES

The high velocities and low trajectories attained by the bullets fired from modern rifles have not been accompanied by corresponding improvement in hitting efficiency, either in warfare or hunting. The ratio of misses to hits in battle is enormous. In war, hunting, manufacturing, transportation and all the activities of life, the demand is for higher efficiency, and from a purely scientific standpoint the new system of rifle sighting described in the following should be of interest. It should be of particular interest to sportsmen, as it probably means that average shooting efficiency can be doubled or tripled. And whatever may be our prejudices regarding war, the fact is that if war is necessary it behooves us to be on the winning side. "The man behind the gun" is the unit of efficiency. If he can be made twice as efficient in battle as the average soldier is to-day, it is equivalent to doubling the size of the army. In the following analysis of the system the advantages are made evident.

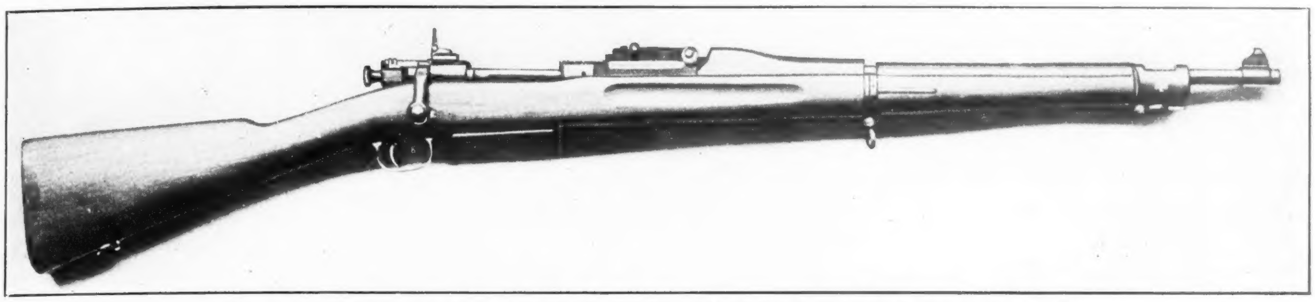

Fig. 1. U. S. Springfield Army Rifle fitted with Remington "Negative Angle" Battle Sight on Bolt

Mr. Harcourt Ommundsen, one of the inventors, by winning the King's prize, the Service Rifle Championship, and Grand Aggregate (the two latter on several occasions), has established his reputation as perhaps the most accomplished rifleman in the British Empire. Mr. Edward J. D. Newitt, who collaborated with Mr. Ommundsen in the practical application of the invention to war and hunting conditions, is equally well known in Great Britain as a marksman, ballistic engineer, and a pioneer of progress in aiming devices for projectile weapons.

The invention should be described as an evolution of the art or method of shooting rather than an improved sight, though sights of special construction are employed in the application of the system. In many respects the invention is analogous to that of the pointed bullet. Though elongated bullets of almost every conceivable shape from cylinders to pyramids have been made, tried and abandoned in the past seventy years, ten years have not yet elapsed since Gleinich demonstrated that an attenuated bullet having a point constituted by shoulders of a specified curvature met with less resistance from the air than any other shape. This enabled the use of lighter bullets without sacrifice of ballistic efficiency; higher velocities thereupon became possible, trajectory was flattened and the efficiency of the rifle, theoretically, was advanced considerably. Whatever may be the theoretical efficiency of rifles as regards ranging power and striking energy, the condition precedent to its realization, either in war or in hunting, is to hit, and for many years past Messrs. Ommundsen and Newitt have devoted their energies to enhancing the hitting efficiency of rifles, this, at present being by far their weakest feature.

When we read of the devastating hurricanes of lead that sweep the modern battlefield and then, by cold mathematics, demonstrate that but one in ten thousand of the bullets constituting this hurricane accomplishes more than to scatter a few grains of mother earth, we may rightly question whether modern war weapons are really more effective than the clubs and bows of our ancestors. It is difficult to realize, but if the statistics of warfare are to be relied upon, it requires twenty-five men to fire two hundred cartridges apiece, or fifty men to fire one hundred cartridges each, to effect one hit in a battle with modern firearms. Indeed, our boasted modern improvements have effected little more than to prolong war, increase the area of the battlefield, the size of armies and expense.

In hunting, the up-to-date sportsman with the best of modern equipments may perhaps kill his game a little further off, but, shot for shot, he can accomplish no more than the aboriginal savage with his bow and spear. It is incontestable that the hitting efficiency of modern weapons has not kept pace with their advancement in other features. Supposing that, by any means, the average of hits to cartridges fired could be extended from one in, say, five thousand to only two in five thousand it would be equivalent to doubling the army; or we might decrease the army by half without sacrificing its fighting strength, and thus economize enormously on time and expense. It is these considerations which have inspired the inventors in their investigation of the causes of the failure of rifles in the field, and their efforts to diminish the difficulty of hitting.

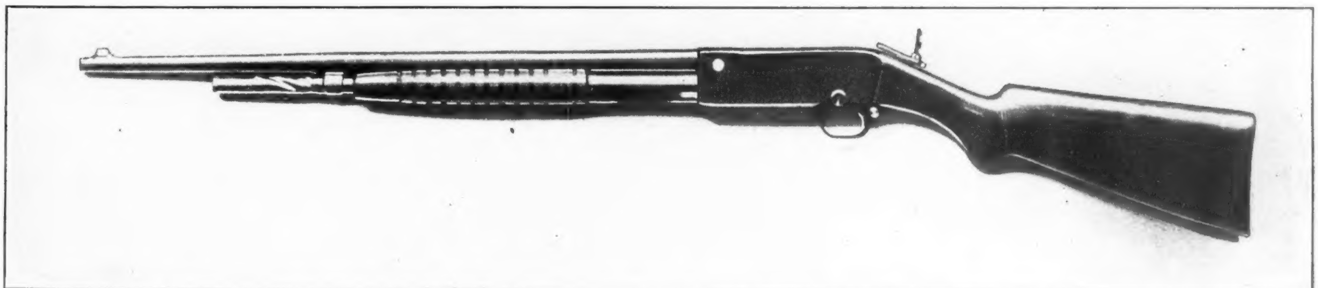

Fig. 2. Remington "Negative Angle" Sporting Rifle

The trouble began with the introduction of the conical bullet and rifled barrel into armies about three-quarters of a century ago. Up to that time a fixed notch for a backsight and a bead or point frontsight served all the requirements of aiming the old smooth-bore muskets—at least as straight as they would shoot. But with the advent of the accurate long-range rifle came the necessity of counteracting the effect of gravity. To get range, the bullet must be kept in the air as long as it takes to perform the journey. If this journey takes one second and gravity has ordained that any unsupported body like a bullet shall fall sixteen feet in one second, then if we desire the bullet to travel to a mark it will require one second to reach, the rifle must be pointed sixteen feet above that mark. But with the old fixed notch and point sighting combination this meant aiming above the object, and incidentally completely hiding it in the process. To overcome this difficulty some

genius devised the tangent elevating backsight, a long name by which is implied mounting the old notch on a slide so that it could be raised and lowered. By this means the axis of the barrel could be fixed at any angle to the line of sight, and aim could be taken at the mark though the barrel was in fact pointing far above it, and by this means this genius and thousands of his descendants have considered that the problem was solved. As a matter of fact, this accomplished very little toward it. It is true that the shooter had no longer to judge how much his bullet would drop in traveling to the mark and then aim an equivalent height above it, but in exchange he had to judge how far the object was away, and adjust the height of his backsight accordingly, and a very little consideration will show that one is precisely the same and just as impossible as the other—in fact nothing had been gained except ability to see the object while aiming.

Very few appreciate how formidable is the problem of hitting an object at any distance with a modern rifle. First, the distance must be ascertained. It might be measured, but as this is seldom possible in hunting or war, it must, of course, be estimated. The rifle requires it to be estimated within a very few yards, and the longer the distance the nearer must be the estimation. No man has yet been or ever will be able to judge distance within the proximity required by the modern rifle sight, for the simple reason that the human body is not

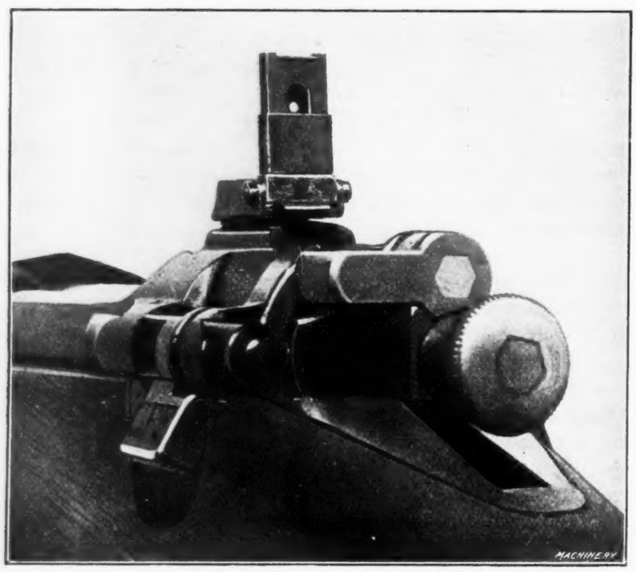

Fig. 3. Close View of Army Rifle Sight

equipped with any distance-measuring faculty and a non-existent faculty cannot be developed by any process of education or training.

But, assuming that this difficulty has been overcome there still remains the necessity of elevating the backsight with an exactitude which is measured by one or two hundredth parts of an inch, a feat very difficult to accomplish under the most favorable circumstances and wholly out of the question in the stress of battle. These difficulties have been to some extent appreciated and something has been accomplished toward ameliorating them—this, curiously enough, by reversion to the old fixed notch backsight.

The U. S. Springfield, and the military small arms of several other nations, is equipped with what is known as a battle sight. This may be a separate fixed notch, or the ordinary elevating notch when in its lowest or normal down position. This fixed notch battle sight has a certain amount of elevation which imparts a line of flight to the bullet in which at no point in the length of the resulting range does the bullet rise higher above the point of aim than the height of the object fired at. If aim is always taken with such a sight at its lowest extremity, the object will be hit in some part at any point in the range. But here another difficulty arises. In battle the usual objects are men who alternately adopt postures of different heights. When advancing they may be erect or about sixty-eight inches high, when halting they will lie down and offer a target anywhere from twelve to eighteen inches high, or they may kneel, crouch or assume positions giving any height between the two extremes. Consequently there is a good deal of difference of opinion as to the most appropriate elevation to adopt.

For example, if the battle sight is elevated to give a trajectory (the curved path of a bullet is called trajectory) of which the vertex is the height of the average erect man, then for the greater portion of its range it will pass harmlessly over the

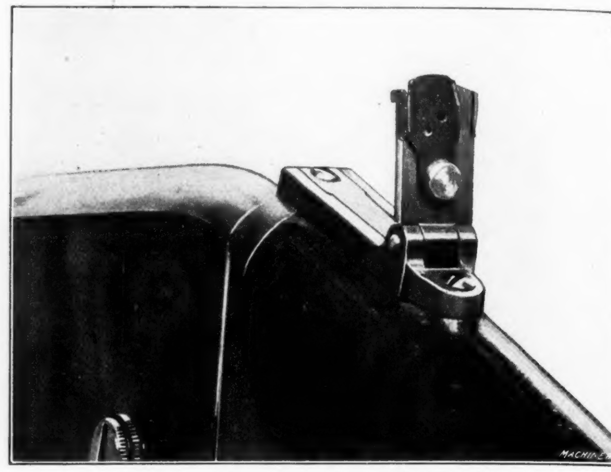

Fig. 4. Close View of the Sporting Sight

heads of men in the kneeling or prone positions. On the other hand, as the higher the trajectory the longer is the range, a sixty-eight-inch trajectory is useful for keeping the enemy at a distance, and it is not yet certain whether or not the most effective results will be obtainable at the relatively longer distances—when the enemy is in an erect position and offers a larger target, or when he is closer and lies down. Most nations have adopted a battle sight elevation which gives a trajectory somewhere between fifteen and twenty inches high at its vertex, thereby sacrificing the longer range possibilities of a trajectory suitable alone for erect men in favor of a trajectory suitable for men in the prone position. Some, on the other hand, have adopted an elevation giving a trajectory of intermediate height.

The U. S. Springfield battle sight, for example, has a trajectory thirty inches high and will hit any object thirty inches and upward in height at any distance up to five hundred and thirty yards, but as a prone man seldom presents a target higher than eighteen inches it is not suitable for shooting at prone men. The selection of a thirty-inch trajectory for a

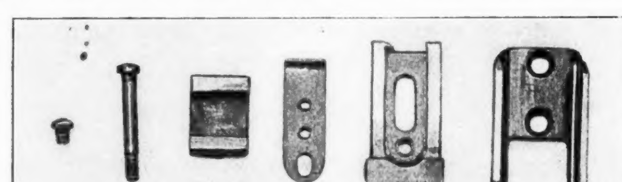

Fig. 5. Component Parts of Army and Sporting Sights

battle sight elevation cannot be described as ideal inasmuch as it has a limited danger zone for men who are in the prone position and unnecessarily shortens the range for men in the erect position. Doubtless the idea is that indifferent shooting on the part of the soldiers will discount this difficulty. It might similarly be argued that an inaccurate rifle might cor-

rect bad aiming, and to continue such an argument to its logical conclusion it might be contended—indeed it already is contended by some distinguished military authorities—that in the hands of the average soldier the best results are obtainable from an improperly elevated, inaccurate rifle, incorrectly aimed. This is tantamount to admitting that hitting on the battlefield is largely a matter of chance and it is better to rely upon chance and probabilities than on skilled men and perfect rifles. Notwithstanding this, most nations are spending vast sums on perfecting their rifles and training their soldiers to shoot, while the possibilities of chance, as a means of attaining fine effect in battle, have been abundantly exploited in recent wars, and have proved a most uneconomical and hopeless failure.

However, such are the difficulties of estimating distance and adjusting sight elevation, that the practicable battle range of military rifles has come to be regarded as the range given by their battle sights, within which both estimation and adjustment are unnecessary, and the trend of development in war rifles in the last ten years has been entirely in the direction of increasing the velocity of the bullet with the one purpose of extending the battle sight range. The present practical limit of velocity of rifle bullets is something less than 3000 feet per second. The German Mauser has about 2900 feet per second velocity; the Swiss rifle, about 2800 feet; the U. S. Springfield about 2700 feet, and the latter is by far the most accurate military small arm extant. To increase the range given by a thirty-inch trajectory by 100 yards, would imply an increase in the velocity of the bullet of about 500 feet per second, which at present is out of the question; consequently, further increase of range from enhanced velocity is unlikely for some time to come. A considerably lengthened range may, however, be obtained by development of the battle sight and the improved method of sighting known as the Remington "negative angle" system. This battle sight by which the system is adapted to military small arms, consists of a plate containing two apertures, mounted in a compact framework and attached to the rifle just over the bolt, conveniently close to the eye, in which position alone the aperture or peephole as an aiming medium is reliable. One aperture gives a trajectory sixty-eight inches high, the other thirty-six inches high. Only one aperture is exposed at a time, the act of exposing one automatically obscuring the other. The change is made by moving the slide up or down between its arresting stops about one-tenth inch. This can be done instantaneously by sense of touch alone without the assistance of the eye and with the rifle at the shoulder.

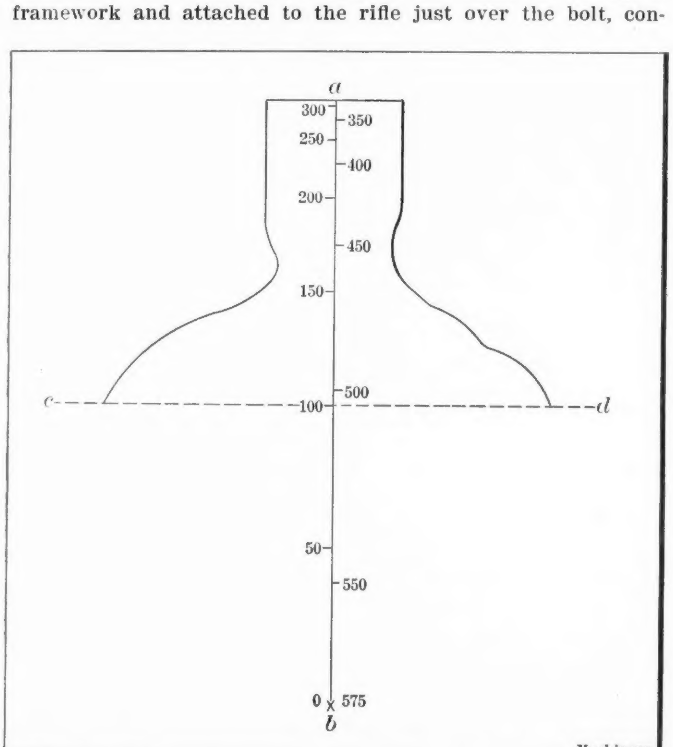

Fig. 6. Illustrating the Relation of Springfield Army Rifle Trajectory to a Prone Figure

It is a generally accepted theory that the trajectory given by any angle of elevation will be constant relative to the line of sight, though the latter may be raised or depressed relative to the ground, within all necessary limits. Messrs. Ommundsen's and Newitt's invention prescribes a practical application of this theory, whereby a trajectory higher than the object can be employed and the range correspondingly extended.

In Fig. 6, a–b is a vertical projection of the thirty-six-inch trajectory of the Springfield rifle, superimposed on the figure of a prone man drawn to the same scale. The horizontal dotted line c–d is the ground line and × the negative point of aim. The figures on each side of a–b indicate the height of the bullet at the distance expressed by these figures. It will be observed that the distance between the ground line and the point of aim is equal to the height of the figure, and it is obvious that such a figure shot at with such a trajectory would be hit at any distance between 90 and 505 yards if aim were taken at ×. Similarly, if the aim were raised to the ground line, any object exceeding thirty-six inches in height would be hit at any distance up to 575 yards. Mainly for the purposes of repelling an attack and keeping the enemy at a greater distance than this, the second aperture in the battle sight is provided. With the slide up, a sixty-eight-inch trajectory, giving a range of about 750 yards, is obtained. With this, by ground line aim, a standing man will be hit at any distance less than 750 yards and by negative aim, kneeling and crouching men are vulnerable up to about 600 yards.

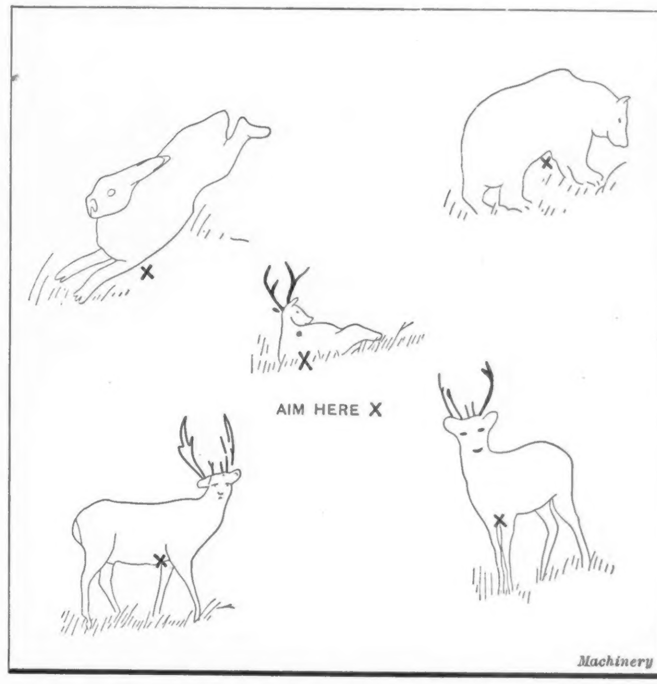

Fig. 7. Showing Constant Points of Aim on Animals to hit Vital Spot

Under the present system if the Springfield rifle were equipped with a battle sight suitable for men in the prone position, *i.e.*, a battle sight giving a trajectory eighteen inches high, the battle range for figures of all heights would be limited to about 450 yards. With the Remington negative angle battle sight, prone men may be hit to 500 yards, kneeling men to 600 yards and erect men to 750 yards, a gain which is not likely to be reached by any enhancement of velocity for many years to come. Moreover, whatever velocity may be reached in the future, a similar relative gain may still be made by means of this sighting system. It will probably be contended that to aim negatively requires as much estimation as to judge a distance. It will not, however, escape observation that as the object is necessarily visible while aim is taken, it furnishes a visible gage with which a depth below equal to its own height is quite easily compared. Nothing, in fact, in the nature of judgment or estimation is required, while the curve of the trajectory affords considerable latitude at all distances for the inevitable errors of aim which occur with all sights and all marksmen under all conditions.

In using the negative angle battle sight, distance and sight adjustment are put entirely out of consideration. When an

object is near enough to be aimed at it can be hit. Three rules alone are employed: 1. At prone men use lower aperture and aim negatively. 2. At erect men use the higher aperture and aim at ground line. 3. At men in any other position use the lower aperture and aim at the ground line.

In applying the system to the conditions of hunting, a somewhat similar sight, giving twelve-inch and five-inch trajectories, is employed. Change from one aperture to the other is made by pushing a disk from left to right. The higher aperture will always be employed for deer, bear, moose and other large game, the lower for woodchuck and small game. The illustration Fig. 7 shows the constant point of aim which must be taken under all circumstances, irrespective of distance. With a modern American rifle, such as is commonly employed for such game, the range given by the hgher aperture is about 300 yards and by the lower aperture about 150 yards.

It will not be forgotten that though a deer is a relatively large animal, the vulnerable portion—the heart region—which must be hit to insure a clean sportsman-like kill is only about six inches diameter, and to hit this even at short distances with the present sights and system of shooting, demands the same correct estimation of distance and exactitude of sight adjustment as was presented in the military problems just discussed. In the majority of instances, misses while hunting are attributed to bad shooting or inaccuracy in the rifle. As a matter of fact ninety per cent at least are due to improper elevation of the backsight. To correctly elevate a backsight, is just as difficult in hunting as in war, and a sight which is independent of adjustment is just as essential in one case as in the other.

The average deer is about eighteen inches thick above the prescribed point of aim, and from 40 to 250 yards is the range given by a twelve-inch trajectory. The path of the bullet then will lie between three inches high or low of the center of the heart region. Thus, an effective sporting range, sufficient for all practical purposes, is obtained.

This system is applicable to rifles of all calibers, and the effective sporting range will be governed by the velocity of the bullet, that of black powder velocities (about 1250 feet per second), being somewhat less than that of smokeless powder velocities (about 2000 feet per second).

The Remington negative angle sighting system described in the foregoing, comprises not only the sights, which in themselves are simple, but the system of aiming, which within limits automatically compensates for the fall of the bullet in its flight. It will be placed on the market by the Remington Arms-Union Metallic Cartridge Co. of Ilion, N. Y. and Bridgeport, Conn.

* * *

PROPOSED CHANGES IN FRENCH PATENT LAW

The French Government is proposing to amend the patent law of France. The main features of the bill, which is likely to pass, are as follows: Duration of patents shall be fifteen years. Samples or models must be deposited in sealed boxes at the same time that the sealed letter containing the application is handed in. Persons resident abroad must appoint a representative living in France who will receive all communications. Applications must be written in French and must not contain any designation of weights or measures other than in the metric system. An inventor of a device already patented abroad can obtain a patent in France. Such a patent will be independent of the patents obtained for the same invention in other countries.

Loss of patent rights will follow (1) if patentee does not pay his annual dues; (2) if the patentee imports into France objects similar to those of his patent, but manufactured abroad; (The loss of patent right will not be incurred by a patentee who can prove that the quantity of goods manufactured in France in the three years preceding action taken for loss of rights has been at least double the quantity imported.) (3) if the patentee has not worked his patent in France or French colonies within three years, or shall cease to work it in any three consecutive years, unless he shows reasonable cause for lack of working.

TESTING LUBRICATING OILS

The *Journal* of the American Society of Mechanical Engineers gives, from a German contemporary, the following practical directions for determining roughly the qualities of lubricating oils without the use of special testing apparatus. To determine the presence of solid impurities in the oil, kerosene is added to half a tumbler of oil until the whole becomes quite thin. The mixture is then passed through filter paper or ordinary white blotting paper, and after all the oil has passed through, the paper is washed with kerosene; the residue on the paper, if any, will show if the oil had any solid impurities. Impurities may also be roughly detected by smearing a piece of writing paper with oil and holding it against the light; if the oil is free from solid impurities the blot will be equally transparent throughout—otherwise, the solid particles will show. The oil must not resinify: to test it in this respect pour it into a shallow dish and leave in a warm place about a week. There must not be the slightest crust at the end of that time. Another way to test it is by mixing it with fumes of nitric acid: if the oil is pure, a thick mass will form in a few hours, while resinifying oil will remain thin. Acids are very injurious impurities in lubricating oil, since in time they attack the machine parts lubricated. To test for them, copper oxide or copper ash is added to the oil in a glass vessel; acid-free oil retains its original color, while, if it contains acid, it becomes greenish or bluish. Another test is to drop the oil on a sheet of copper or brass and leave it there for a week; if the oil contains acid there will be a green spot on the metal. Blue litmus paper dipped in acid-free oil stays blue, but becomes red if the oil contains acid. A good oil must be greasy in order to have good lubricating qualities; to find which of several oils is best in this respect, place a few drops of the different oils on a smooth, slightly inclined metal or glass sheet; the drop of the best oil will travel furthest in a given time.

* * *

GRAND CENTRAL TERMINAL

The Grand Central Terminal which has been in process of construction in New York City for several years, was opened to the public February 2. The terminal is for the use of the New York Central & Hudson River R. R. and the New York, New Haven & Hartford R. R. The main structure is 301 feet wide and 722 feet long at the street level and about 455 feet wide below the street. The tracks are on two levels and are over 33 miles long. There are 42 tracks for through trains and 25 for local trains. The combined capacity of all tracks is over 1000 passenger cars. The total area of the terminal property is about 70 acres. The main waiting room on the out-going track level is 301 feet long, 125 feet wide and 125 feet high. The depression of the tracks and the substitution of electric locomotives for steam locomotives made possible the reclamation of a large area of very valuable land in the heart of the city. The terminal property comprises, besides the main terminal structure, warehouses, the Grand Central Palace, the Biltmore Hotel and other structures. The streets formerly cut off by the railway have been restored to the public and a commercial center established convenient to shipping facilities. The capacity of the new terminal is estimated to be 100,000,000 people a year.

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EFFECT OF POOR ILLUMINATION ON THE EYES

Experiments undertaken with the object of ascertaining the effect on the eye of various kinds of illumination, indicate that after three hours' work in ordinary daylight, there is hardly any change in the working efficiency of the eye. After three hours' work in artificial light consisting of incandescent lamps with flat porcelain reflectors over them at the height of about eight feet from the floor, the keenness of the eyesight had fallen off very greatly, and a distinct loss in the muscular adjustment for accurate vision was perceptible. The study of the proper artificial lighting is a subject of much importance, and it is likely that the next few years will see a still greater improvement with regard to artificial illumination than has the last decade.

CROWN FACE PULLEYS*

A REVIEW OF THE PRACTICE IN PULLEY CROWNING AND SUGGESTIONS FOR OBTAINING THE BEST RESULTS

BY GEORGE N. VAN DERHOEF†

The crowning of pulleys is defined by one of the well-known dictionaries, as near as I can remember, somewhat as follows: "Pulleys are commonly made with a rounded face because the convexity keeps the belt on when various other things make it run off." This definition may, perhaps, be considered by some of the readers quite as lucid and as much to the point as the following dissertation, but as the subject of pulley crowning has received scant attention in the mechanical journals, I will take the risk. I do not know to whom we are indebted for the valuable discovery that crowning a pulley will tend to keep the belt from running off. A number of writers have given, or attempted to give, an explanation as to why a belt runs to the center of a pulley when provided with a rounded face, but I have been unable to find any definite statement as to when or where the practice originated. My impression is that it was the invention or rather discovery of a mechanic employed in an English cotton mill.

The amount of crowning that should be provided varies according to different writers; hardly any two make the same recommendation. Some writers give the amount of crown as a percentage of the belt width, and give figures all the way from 1/12 to 1/96. Others give it as a taper of so much to the foot, quoting 1/16 inch, 3/32 inch, 1/8 inch, 3/16 inch, 1/4 inch, 5/16 inch, 3/8 inch and 1/2 inch. One writer seems to have a leaning towards as much as 1 inch taper per foot. In addition to these simple standards, some express a liking for fancy curves or for faces made truly rounded, using a radius of from three to five times the belt width or more.

Some authorities, however, seem to have realized that there are several factors entering into the question of pulley crowning and have tried to make allowance for them. For example, one recommends a crown of 1/20 the width in the case of leather belting and 1/150 the width for cotton belting; another allows 1/16 to 1/8 inch per foot of width for high speeds and 1/4 inch for low speeds, while another recommends not over 1/8 inch for 2 feet breadth for pulleys 48 inches in diameter or less, and in no case over 3/16 inch. Undoubtedly this wide divergence in recommended amounts of crowning is based on the judgment and observation of the individual writers or in some cases on shop practice handed down from a rather nebulous past. Very little in this mass of data appears to be based on any definite theory as to what should be used, or to have been checked up by actual experience.

The general theory as to why a belt runs to the center of a crown face pulley is simple, but it is not altogether easy to explain without a number of diagrams, which might after all, make the matter confused. In the first place, a crown face pulley should be considered as two tapered cone pulleys having the large ends placed together. We can then think of the belt being divided in the middle or as two belts running side by side, each on its own taper cone pulley. Now we know that if we have two parallel shafts, one which we will call the driving shaft carrying a straight face pulley and the other a taper cone pulley, and the two pulleys are connected with a belt, the belt will travel toward the large end of the taper cone pulley when started into motion. There are two reasons for this: First, if the belt leads on to the taper cone pulley exactly at right angles to the shaft, instead of going around the cone in its own plane, it tends to describe a conical spiral path on the face of the cone, the spiral running toward the large end of the cone like a screw thread. This can be proved by simple experiments which anyone can make. Take a piece of paper three or four inches wide and about eight inches long; roll this up into a cone so that one end is about three-fourths the diameter of the other, and fasten the edges together with paste. Then trim both ends square with the scissors. Now mark a straight line from end to end of the cone and square with the ends; draw an arrow near the small end of the cone and at right angles to the line and with the point of the arrow just touching the line. Next, cut the cone from end to end just behind the arrow and lay the paper out flat on the table. This will constitute what a draftsman would call "a development of a conical surface on a plane." With a rule, extend the arrow all the way across the paper and you will find that it goes across in a diagonal direction. Now roll the paper up again and paste the edges together and you will see that this straight line becomes a screw thread running toward the large end of the cone and representing the path the belt tends to take.

The second reason referred to is this: On account of the cone being larger at one end than the other, the edge of the belt nearer the large end is longer than the edge nearer the small end; this makes the belt curve or bend sideways slightly. The curvature of the belt is towards the high side and it leads the belt onto the cone at a slight angle instead of square with the face. As everyone knows, when a belt is led onto a pulley at an angle, it runs to one side. We see from this that there are two factors acting at the same time, both tending to make the belt travel toward the large end of the cone. This dual action is very important and is commonly overlooked; it enables us to explain some of the queer things that happen to belting in the shop.

In the foregoing, the supposition was, of course, that the shafts are parallel. Now take a case where the shafts are level and at the same height from the floor but not parallel. Arrange the shafts so that they are closer at the small ends of the cones than at the large ends. If the lack of parallelism is just right, the belt will run to the small end of the cone instead of the large end when started in motion. The reason for this action is that the angle due to curvature sideways of the belt as previously explained, is not as great as the angle the other way, so that the belt is held onto the taper cone because of the lack of parallelism of the shafts. The result is that the tendency to run to the side from which the belt leads is greater than the tendency to run to the large end of the cone and the belt actually travels to the small end. If the angle between the shafts is just right, it is possible to run a belt on the middle of the face of a taper cone pulley.

To go back to the original proposition of two taper cones with their large ends placed together and the belt split down the middle so as to form two belts side by side, it could be explained that as each belt runs to the high side of its own taper cone, both belts will run together at the middle where the tendency of one of the belts to run to the right will be opposed by that of the other to run to the left. This is why it is so common to see a belt running first a little to one side and then to the other. Other things being equal, the force with which a belt runs to the large end of a cone depends on the width, so that in the case of a belt running on the crown face pulley the moment it gets out of center a greater force on one side than on the other tends immediately to force it back to its proper position. When the shafts are perfectly straight and in line, the pulleys accurately finished, and the belt uniform in width, thickness and character of surface, the side motion will be so small that it can hardly be seen. As all these conditions are seldom realized, it is common practice for pulleys to be made slightly wider than the belt.

There is sometimes a belt in the shop which persists in running to one side of a pulley until about one-half inch hangs over one side and it continues to run this way steadily. The reason for this is that one of the conditions is present under which a belt tends to run to the small end of the cone. The belt runs over the edge of the pulley an amount sufficient to relieve the tension on the over-hanging edge. The balance of forces then being adjusted, the natural tendency of the belt to run to the large end of the cone holds the belt from going further toward the small end. The common practice in the shop, of knocking a countershaft out of line in order to make the belt run right, is perfectly justifiable when the pulleys or

* See "Pulley Crowning" by George A. Gauthier in the September, 1905, number.

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belts are imperfect, unless the more scientific procedure is adopted of throwing them away and buying first-class equipment.

It is not generally known that straight face pulleys are often in effect cones of a very slight taper. This may be due to hard spots in the face which vary the coefficient of friction and make the pulley act as though it were turned with a slight taper. The actual amount of crown that should be given a pulley depends on the "other things" referred to in the dictionary definition. Among these may be cited: lack of parallelism between shafts; shafts that are not straight; pulleys that are not turned true or that are finished smoother on one side than on the other; springing of shafts under changes of load; belts that are not uniform in quality or in thickness, or which do not stretch uniformly when under strain. Lack of balance of pulleys is probably not a direct cause, but doubtless is an indirect cause as it produces some of the direct causes mentioned.

The wide variation in the method of crowning recommended by various authorities, can now be understood. In those industries where high-grade machinery is used and the best attention is given to keeping shafting in line as well as to the careful selection and care of belting, a tendency toward the use of small amounts of crowning will be observed.

Inasmuch as the circumference of a pulley is greater at the center of the crown than at the edges, it is obvious that a small slip and consequent loss of power is taking place all the time the crown pulley is in operation. This is one of the reasons, particularly in the case of large powers, why rope driving is so extensively used. If a pulley has a 1/4-inch crown, it means a difference in the circumference of a little more than 3/4 inch. In a revolution of the pulley, this difference must be taken out as slip. If the speed of the belt is a mean between the two circumferences, there must be forward slip to part of the belt and backward slip to the remainder. The amount the slip is modified by the change in length of the belt as it stretches in going from the slack side of the drive to the tight side and shortens in going from the tight to the slack side, is too slight to make material difference. The 3/4 inch just mentioned, in the case of a 4-foot pulley, is distributed over slightly more than twelve feet of the circumference, while with a 12-inch pulley it is distributed over slightly more than three feet. If it were not for other reasons, the crown should be proportional to the diameter of the pulley. As it is, we can confidently expect better wear of a belt and higher efficiency of the drive, by using large pulleys than small ones. We can also see one of the reasons why belts to high-speed small pulleys should be made amply wide so as to be under small strain per inch of width.

The crowning of a pulley only tends to keep the belt on when the belt as a whole does not slip. A slipping belt will run off a crown faced pulley quicker than from a straight faced one. Where the danger of slip is great, particularly in the case of a small pulley driven by a large one, it is advisable to use a moderate crown and provide flanges. The moderate crowning will lessen the danger of slip and the flanges will keep the belt on where a high crown will not. Do not infer from this that flanges are better than a crown—as a rule they are not—but simply regard them as an expedient or safeguard.

The amount of crowning that can be safely used depends upon the resilience of the belt. When the stretching of the belt due to the crown exceeds the elastic limit a permanent set is given to the belt which detracts from or neutralizes the action of the crown. This explains the reason why pulleys on which cotton and certain other kinds of belting run, are given a much smaller amount of crown than that which can be safely provided for leather belts. In this connection, the difference between stretching and elasticity should be fully understood. Stretching may be described as permanent elongation of the belt due to stress, while elasticity is the amount the belt lengthens under stress but which is taken up again as soon as the pull is removed. We try to buy belts of a high degree of elasticity, but we stretch them thoroughly before we put them in use.

It is claimed that a leather belt may lead onto a pulley at an angle of 2½ degrees when the pulley has a crown of 1/20 the width of face. In other words, this would mean that an error in alignment of the shafts of 2½ degrees is permissible.

The recommendation has been made that one of the pulleys of a pair be made with a straight face so that only one of the pair of pulleys would have a crown. This writer considers it to be pernicious advice, inasmuch as the belt would be less injured by a crown of one-half the height on each pulley. When a belt is used to drive a pair of tight-and-loose pulleys, it is practically necessary to make the driving pulleys with a straight face so that the belt can be easily led back and forth across the face as it is shifted from the loose pulley to the tight pulley, or *vice versa*. The tight-and-loose pulleys, however, are made with crowning faces so as to hold the belt in place after it is shifted. The belt shifter is simply a device used to upset the balance of forces described and that hold the belt to the center of the pulley so that it will run over onto the other pulley. The shifter is for the purpose of setting up a condition under which the natural forces will lead the belt to the desired position.

Now while the foregoing mostly applies to the belt as it runs onto one of the pulleys, it is, of course, obvious that the same action takes place when the belt returns from the second to the first pulley. When the second pulley only is crowned, the belt is delivered to the first pulley at a slight angle but it has a tendency on this pulley to travel to the same side as it does on the crown face pulley. It was probably this fact that led to the recommendation of making one of a pair of pulleys with a straight face. The plan, however, of giving the smallest possible crown to both pulleys, certainly seems to be based on the better principle and is the one largely followed.

The question as to whether the crown should be formed by a curve or made flat on each side of the center line like two taper cones joined at their large ends, is open to discussion, but with the slight amount of crown that should be used, it is practically impossible to determine which is the better, so it may properly be considered as a manufacturing detail.

* * *

LOHMANNIZING

The protection of iron and steel is at present effected by means of three zinc-coating processes: the old hot-galvanizing process, the cold-galvanizing process (electrolytic zinc plating) and the sherardizing process. The process of "Lohmannizing," invented by Mr. H. J. Lohmann, and described in the journal of *Engineering Chemistry*, differs from these in that it is not restricted to the application of zinc coatings, but may, it is claimed, be used for coating of zinc, lead, and tin in varying preparations to suit the requirements of each case.

The process of Lohmannizing begins, as is usual, with pickling the iron and steel articles in a bath of sulphuric acid. Then the article is dipped into the Lohmann bath, which, being "composed of an acid and an amalgamated salt, further cleanses the pores and cavities, and deposits metallic salt upon the entire surface, penetrating into the most minute pores and cavities." The Lohmann bath is a solution of hydrochloric acid, mercuric chloride, and ammonium chloride. Next follows drying, and then immersion in the molten protective alloy, the temperature of which is 950 to 1000 degrees F. An amalgam or chemical union is then formed, it is claimed, between the amalgamating salt and the protective alloy. The temperature of the immersed article rises as it comes in contact with the molten alloy, and, when it reaches a temperature of 500 degrees, volatilization of the mercury occurs.

Since the mercury passes from the surface of the alloy, the complete surface is said to be left free and open for the protective alloy to fill its pores and cavities, there being freedom from oxidizing influences. Inasmuch as the claim is made that a perfectly satisfactory lead-plating can be obtained by the Lohmann process, it would seem that this process would be of value in the production of lead-lined apparatus.

* * *

Aluminum is now extensively used in galvanizing to produce a smooth, bright coating on the galvanized articles. A very small quantity, says the *Brass World*, is required, and is added to the zinc when melted.

FIXTURES AND GAGES USED IN MANUFACTURING THE "MULTIGRAPH"

DEVICES USED IN THE LATHE, SHAPER AND MILLING MACHINE—METHODS OF INSPECTION

BY DOUGLAS T. HAMILTON*

The "Multigraph," manufactured by the American Multigraph Co., Cleveland, Ohio, is an office appliance used principally for producing typewritten form-letters and circulars in quantities. The device is a printing press on a small scale, and can be used for a large range of work usually produced on a much larger and costlier machine. It consists of two drums; one of these, called the "supply drum," carries the type, from which it is supplied to the other drum, called the "printing drum"; these drums are mounted end to end on a base that may be set on a table. A wide ribbon or printer's ink attachment is used in printing, the ribbon being used largely for form-letter typewriting, and the ink attachment chiefly for circulars. There are a large number of interesting fixtures and gages employed in the manufacture of this office appliance, and a few of these will be described in the following.

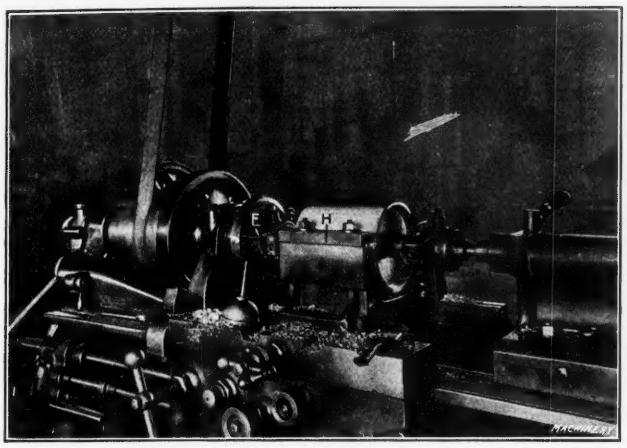

Fig. 1. Counterboring Fixture for Segmental Type Bar Drums with Work in Place

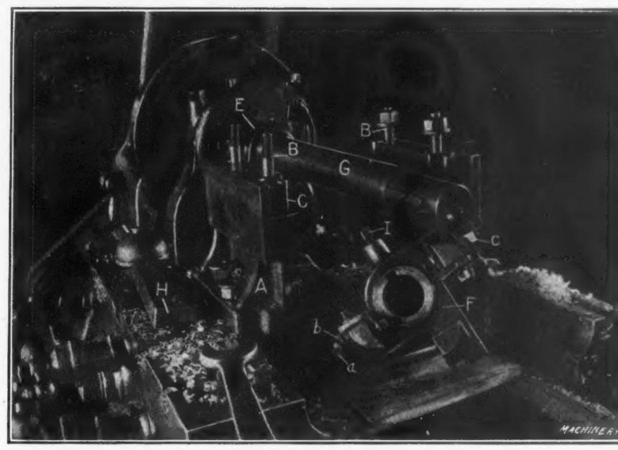

Fig. 2. Fixture shown in Fig. 1, with Work and One Tool-head removed

Fig. 3. Truck carrying Segmental Drums representing Eight Hours' Work in the Fixture shown in Figs. 1 and 2

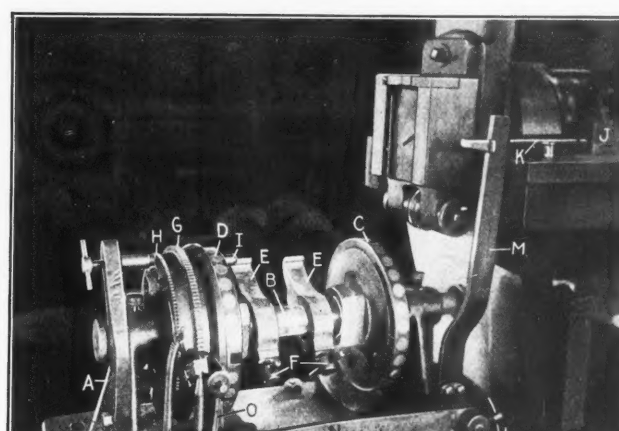

Fig. 4. Fixture shown in Fig. 5, with Work removed, showing Method of clamping and locating the Segments

A Counterboring Fixture for the Lathe

The drum which carries the type is made in two pieces from aluminum castings, see Fig. 3. The first machining operation on these aluminum segments is to surface the faces which are to come in contact. This is a simple operation which is performed on a disk grinder. The next operation is to counterbore both ends and face the drum to length. A special fixture applied to an 18-inch Bradford lathe, as shown in Figs. 1 and 2, is used for this purpose. As these illustrations show, the regular tool-slide has been removed and a cast-iron fixture *A* substituted. This casting is machined to fit the dovetail on the top of the carriage, and is clamped rigidly to it. The top portion of the casting is machined to form a seat for the aluminum drum segments, and is drilled and tapped for studs *B*. Dowels *C* locate the segments endwise, by engaging in slots in lugs cast on the segments.

The machining is done with tools held in the two tool-heads *E* and *F*, fitted to arbor *G*, which is provided with a tapered shank and held in the lathe spindle. The tool-heads are driven by the arbor by means of keys which are a sliding fit in the heads. Three high-speed steel cutters are held by clamps in each tool-head; cutter *a* is used for facing the castings to length; cutter *b* machines surface *K*, Fig. 1, and the bottom of the counterbore, while cutter *c* rounds the inner corner.

The operator first places the two aluminum segments in the fixture, face to face, locating them endwise by the slots in the lugs; then he places the two clamping strips *H*, Fig. 2, on top of the castings and clamps them in place. Tool-head *F* is now slipped onto arbor *G* where it is clamped by set-screw *I*, bearing on the key. The tailstock is brought up and clamped, and the center located in the center in the end of the arbor. The lathe is then started and the fixture moved towards the faceplate, bringing the castings in contact with the tools in tool-head *E*. The tools are fed by hand, and the carriage comes in contact with a stop on the bed of the machine when the cutters have reached the proper depth. The carriage is now moved toward the tailstock, bringing the castings in contact with the tools in tool-head *F*, which perform the same operations on the other end of the castings.

The largest diameter of the counterbored portion in these segments is 7 inches and the depth to the shoulder ¾ inch. Kerosene is used as a cutting compound and is applied with a brush to the cutters. A very smooth surface is produced. The value of the fixture, from a production standpoint, is well brought out by the fact that about fifty pairs, or one hundred of these castings, are machined in 8 hours. This number is shown on the truck in Fig. 3, which is designed for holding a day's work.

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Shaper Fixture for "Rolling-in" the Type-retaining Rails

After the aluminum segments composing the printing drum have been faced to length and counterbored, and after a few minor operations have been performed, they are taken to a Brown & Sharpe automatic gear-cutter, where the slots for the type-retaining rails are cut and the outside is machined to the proper diameter. Three milling cutters are used; the one in the center cuts the slot, and the two outer ones finish the outside circumference. The next operation, after the rails for holding the type have been put in place, is to roll the metal between the rails or bars, so as to bind them securely in the segmental drum.

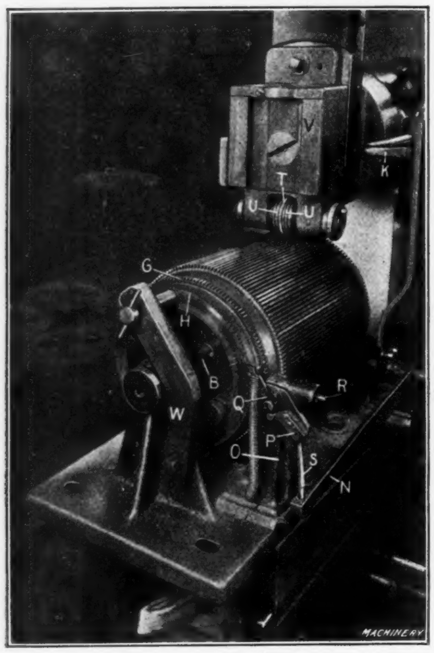

Fig. 5. Shaper Fixture for "rolling-in" Type Bar Rails on Segmental Drums

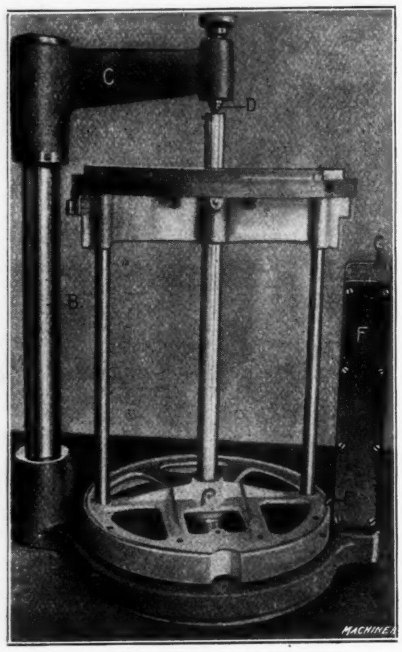

Fig. 6. Gage for testing Segmental Drum Frames

The "rolling-in" operation is done in a crank shaper as shown in Figs. 4 and 5. The fixture *A* is clamped to the table of the shaper. It is provided with two arms which act as bearings for shaft *B*. This shaft carries two disks *C* and *D* on which one segment of the drum is held by the hook clamps *E*, which grip a rod held in a drilled hole in the webs on the inside of the drum casting. The hole in the hook clamps is considerably larger in diameter than shaft *B*, so that when the thumb-screws *F* are screwed down on the shaft, they draw down the hooked ends of the clamps which grip the rod passing through the webs in the casting, thus holding the latter securely on the two disks.

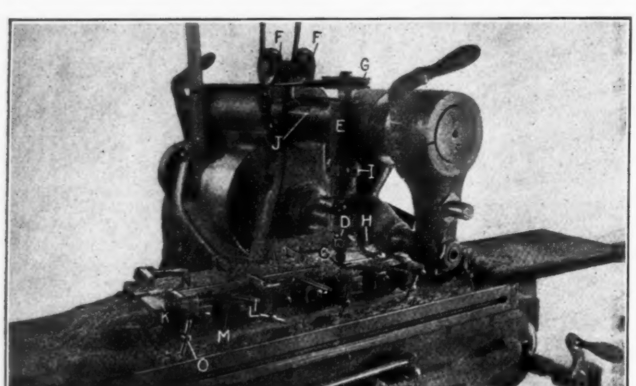

Fig. 7. Special Dovetail Milling Attachment for Front Cross-bar of Character Board

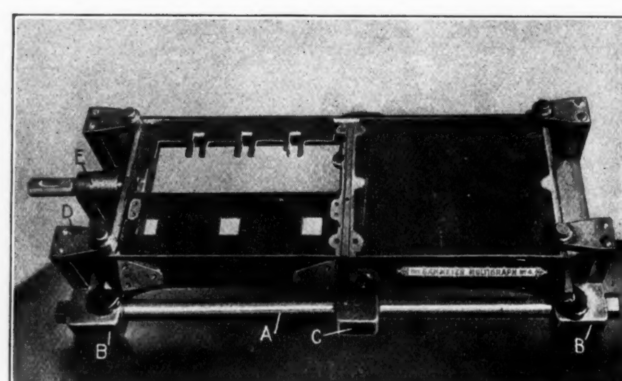

Fig. 8. Gage for testing Milled Surfaces and Drilled Hole in the "Multigraph" Base

An indexing ring *G* having 144 teeth is attached to disk *D* by two studs and thumb-nuts. This ring is turned down on one end and the ratchet ring *H* is screwed and doweled to it. A hole is provided in the segment which, when it is placed in the fixture, locates it in the proper position by means of pin *I*.

The indexing mechanism of this fixture is rather novel. Two studs *J*, only one of which is shown, are screwed into the flange of the ram. These carry bent rods *K*, which are adjustable and held in place by set-screws. On a machined seat of the fixture is held a small bracket *L*, provided with two arms that act as bearings for a fulcrum pin. This pin carries a short two-armed lever, which is connected with levers *M* and *N*. On the front end of the fixture is held another bracket *O* which carries the locking pin and the indexing mechanism.

The indexing mechanism, see Fig. 5, consists of a plate *P* which is fulcrumed on a shouldered screw in bracket *O* and carries ratchet pawl *Q*, kept in contact with the ratchet disk by means of a close-wound coil spring. Plate *P* is also provided with an eccentric slot in which a pin driven into locking pin *R*, and working in an elongated slot in the boss of the bracket, operates. This pin controls the outward movement of a locking pin which is pushed inward by a coil spring. Plate *P* is connected to lever *N* by link *S*.

On the back-stroke of the ram of the shaper, hooked rod *K* comes in contact with lever *M*, pulling it back, and raising the forward end of lever *N*. This lever, through the medium of link *S*, pushes up plate *P*, and as this is fulcrumed by a screw to the bracket, the pawl which is held to it also moves upward, rotating ratchet disk *H*. Previous to rotating the ratchet disk, the eccentric slot in plate *P* acts on the pin fitting in it, which, as the plate is raised, withdraws the locking pin *R* from the teeth in the indexing ring *G*, allowing the disks carrying the work to be rotated through a circumferential space equal to one tooth.

The tool used in "rolling-in" the rails consists of a knurling ring *T* with two spacing rolls *U* on each side of it. These rolls are held on eccentric sleeves on an arbor which passes through the lugs on the holder *V*, substituted for the regular clapper-box holder. The reason for holding the rolls on each side of the knurl on eccentric sleeves, is to provide for adjustment, so that the rails will always be held down tightly against the bottom of the slots in the drum segments. The knurled nuts on each side of the lugs are used to change the throw of these rolls; the nuts are held by screws when located in the proper position. The arm *W* attached to the forward end of the fixture is provided with a screw which takes the thrust of the "rolling in" tool.

A Dovetail Slot Milling Attachment for a Plain Milling Machine

An interesting fixture applied to a plain milling machine for milling a dovetail slot in the front cross-bar for the character board used on the "Multigraph," is shown in Fig. 7. The additional milling arm, which is applied in a rather unique manner, is used to carry and operate a dovetailing cutter, thus making it possible to completely mill the groove in one operation. The milling cutter used in roughing out the slot to the exact depth and width at the top is held on the regular arbor, and produces a slot in the brass bar as shown at *A* in Fig. 9. The dovetailing cutter *C*, shown in detail in Fig. 9, is held in a vertical spindle *D*, Fig. 7, which is rotated in bracket *E* by a round belt driven

from the countershaft and running over idler pulleys F to pulley G on spindle D.

Bracket E, as shown in Fig. 7, fits over the overhanging arm of the milling machine and is also supported by the work arbor. The lower arm of the bracket is split, and is provided with a clamping bolt, so that the lower bearing can be adjusted to compensate for wear. The vertical adjustment is effected by graduated nut I. The upper portion of the bracket is L-shaped and carries a rod J to which the arm holding the idler pulleys is held. The milling cutter H precedes the dovetailing cutter, and thus relieves the latter from the heavy cutting.

The cutter used for finishing this dovetail slot is shown in detail in Fig. 9. It consists of a cutter head and shank made in one piece, and is drilled and tapped at the front end for the adjusting screw a, the head of which is tapered to an included angle of 15 degrees. The cutter has six teeth and three slots extending part way up the body, so that it can be expanded by the screw a when adjustment is necessary on account

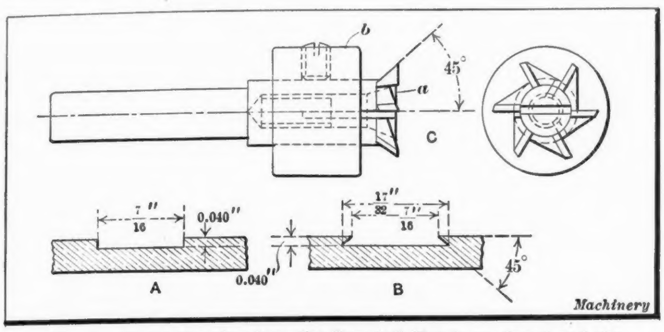

Fig. 9. Views showing how the Dovetail Slot is produced and the Dovetail Cutter used

of wear or when the cutter has been ground. A collar b, held on the body of the cutter by a headless screw, supports the three "cutting members" when in operation on the work.

The fixture used in holding the brass bar when milling the slot in it consists of an iron casting K bolted to the platen of the milling machine. This base casting is provided with a machined ledge on its top face against which the brass bar is held by four eccentric clamps L. These clamps are held on brackets M, bolted to the side of the fixture. The bar is held down on the fixture with five bell-crank clamps N also held on small brackets bolted to the fixture, and provided with "heel" thumb-screws O which operate the clamps. The other side of the fixture is similarly arranged to hold the strip for a second operation, which consists in cutting a saw slot for about half its length. These brass bars, which are 20 5/16 inches long, 1⅜ inch wide, and 5/32 inch thick, have the dovetailed slots milled in them at the rate of 20 per hour.

Gage for Assembled Segmental Frame

The frame on which the segmental printing drum is held is tested for truth after assembling, in the gage shown in Fig. 6. The lower part of this gage is an iron plate provided with a boss for holding the upright B; this latter is provided with a head and washer and carries the swinging arm C which retains the spring center D. A solid bushing is located in the base of the fixture, and when the truth of the frame is being tested, the shaft of the frame is located between the center and the bushing. The needle E, held in the box frame F, is bent, as shown, and bears on the machined surface of the frame as the latter is rotated. The arms of the needle are in the ratio of about 1 to 10; the indications are read on the graduated arc at G. Arm C is swung to one side to insert and remove the work.

Milling and Drilling Gage for "Multigraph" Frame

A simple but accurate gage for testing the milled surfaces and the location of the hole for the platen shaft in the base for the "Multigraph," is shown in Fig. 8. The frame of the gage consists of two binding rods A and three cross bars B and C, the outer bars being held against shoulders on the rods by nuts. Fastened to the two outer bars by screws and dowels, are four brackets D, in which flush pins are held for testing the height of the milled surfaces. These pins are kept down by coil springs, and when the casting is milled to the correct thickness, they come flush with the top faces of the bushings inserted in the brackets. If the casting is milled thinner or thicker than the required dimensions the variation is noted by the height of the flush pins in relation to the top faces of the bushings. The hole for the platen shaft is tested by an ordinary plug gage which fits in a bushing in bracket E. The simplicity of construction of this fixture is its especially commendable feature.

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THE ELECTRIC STEEL FURNACE IN FOUNDRY PRACTICE

According to *Metallurgical and Chemical Engineering*, the use of the electric steel furnace in foundry practice is steadily increasing. This type of furnace is now being substituted in some plants for the crucible furnace or small converter, this substitution being due to the possibility of turning out a high quality steel at a much lower cost than is possible with the other methods referred to. The electric steel furnace is especially suited for the production of castings, since its use enables the finished steel to be held in the furnace without changing in composition; in this respect the use of the electric furnace is similar to that of a metal mixer. The temperature of the metal can also be raised or lowered to meet the requirements of different sized castings.

In the Girod electric furnace the hearth consists of a rectangular or circular iron shell which is lined with refractory material. The furnace is fitted with a sliding door through which the metal is charged and the slag removed; the tap-hole is at the opposite end of the furnace from the charging door. The furnace is supported by a steel frame mounted on trunnions so that it can be tipped forward to draw metal from the tap-hole or backward to skim the slag off the bath through the charging door. The furnace can be operated by any system of current—either continuous, single-phase or three-phase —and the usual tension is 65 volts for small furnaces and 70 volts for larger sizes. The current is supplied through carbon electrodes which hang vertically into the melting chamber, their lower ends being near the slag level. At the bottom of the hearth, as far as possible from the upper carbon electrodes, there are six steel electrodes which pass through the refractory bottom, and are in direct connection with the furnace shell and the low tension side of the power plant. By this arrangement current is forced to pass through the charge so that the entire mass is uniformly heated instead of having the bath merely "licked" on the surface by the arc. A further advantage of having current pass through the bath is that an electromagnetic field is produced which gives the bath a rotary movement that accelerates chemical reaction in the steel.

Among recent applications of the electric steel furnace, the following may be mentioned: The well-known German firm of Krupp & Co. has a fifteen-ton Girod furnace which is being used in the manufacture of special steels for war material, fine steels, etc.; this company is also installing a five-ton furnace to be used for melting ferro-manganese, for which purpose it possesses the advantage of reducing the quantity of manganese lost in the process. In France, several electric furnaces are being used for the manufacture of high-grade steel for projectiles, special steels, tool steel and high-grade steel castings. To bring this matter closer home, it may be mentioned that the following American firms are using Girod electric furnaces: the Bethlehem Steel Co., South Bethlehem, Pa., the Simmons Mfg. Co., Lockport, N. Y., and the Washington Iron Works, Seattle, Wash. The Bethlehem Steel Co. is installing a ten-ton furnace and the Washington Iron Works is installing a three-ton furnace which is to be used exclusively in the production of high-grade steel castings.

* * *

It is estimated that about 2400 men will be required for the operation and maintenance of the Panama canal and the departments in connection with it in the Canal Zone. It has been estimated that the annual payroll will amount to about $2,825,000 and that the total operating expense will amount to about $3,500,000 a year. Adding to this the interest on the investment, it is easy to understand that a considerable paying tonnage will have to pass through the canal in order to make it a financially sound proposition.

DESIGNING STEEL TOWERS FOR WOOD TANKS—4

THE FLOOR SYSTEM FOR FOUR-POST TOWERS

BY EDMUND B. LA SALLE*

In the January, February and March issues of MACHINERY, the methods used in designing steel towers for supporting wood tanks were considered in detail. To make a complete structure it is necessary to provide a steel floor system for supporting the tank on top of the tower, and in this article we will carry out the method of designing it. The design of towers sometimes suffers as a result of competition in prices. In structures of any kind where lack of safety—if such should exist—endangers human life, the first consideration should be that of safety. In this article, only the engineering requirements have received consideration, the ability to meet competition being left to the individual designer. The floor system has to support everything but the tower, and in determining the loads on the beams composing the floor system we must know the weight of all the items that it has to carry. In the February issue, formulas were given for arriving at the weights of all parts of the structure, and we will take a 20 foot by 20 foot tank as an example. The rated or commercial capacity of that size tank is 40,000 gallons and the gross capacity, as given in the table of capacities published in the February issue, is 41,540 gallons. In speaking of the size of a tank in gallons—as 40,000, 50,000, etc.—the rated capacity is meant. The total capacity is the one that we use in designing, except in the formulas for finding the weight of towers, but there we use the rated capacity as the formulas were derived upon that basis. The rated capacities increase and decrease by 5000 gallons, and the weights for the various items that are required in designing the floor system were previously found to be as follows:

Water	= 346,167 pounds
Roof	= 4800 pounds
Tank	= 19,200 pounds
Railing	= 440 pounds
Balcony	= 2000 pounds
Floor	= 8308 pounds.
Total	380,915 pounds

The load carried by the entire floor system is seen to be 380,915 pounds. In order to obtain the load on each beam we first compute the load per square foot of area of the tank bottom, and from that we can get the load per lineal foot of floor beam. The floor beams are also called "joists" and we will hereafter call them by that name. The area of a 20-foot circle is 314.16 square feet.

$$\frac{380{,}915}{314.16} = 1212.5 \text{ pounds} = \text{load per square foot of tank bottom.}$$

As the loads on the beams should always include the weights of the beams themselves, we have included the weight of the floor system, but in figuring the loads on the joists we should omit the weight of the cap-beams as they are not carried by the joists. In the February number, the writer gave the following formula for finding the approximate bending moment in the cap-beams:

$$M = \text{diameter} \times \text{capacity} \times 2,$$

where M is the bending moment in inch-pounds. For the example under consideration, we would have:

$$M = 20 \times 41{,}540 \times 2 = 1{,}661{,}600 \text{ inch-pounds}$$

$$\frac{1{,}661{,}600}{12{,}500} = 132.9 = \text{section modulus of required beam.}$$

The nearest size beam to that is a 20-inch, 75-pound I-beam which has a section modulus of 126.9. This would raise the fiber stress to

$$\frac{132.9}{126.9} \times 12{,}500 = 13{,}100 \text{ pounds per square inch.}$$

A fiber stress of 16,000 pounds per square inch is customary, and for such conditions, a 20-inch, 65-pound I-beam would be quite satisfactory. However, that stress is too high as recent tests on I-beams show that they fail at from 30,000 to 35,000 pounds per square inch—or practically at the elastic limit of the material. The writer has made it a point to never exceed a stress of 14,000 pounds per square inch for the cap-beams and 13,000 pounds per square inch for the joists. The customary cap-beam for a 20 foot by 20 foot tank is a 20-inch, 65-pound I-beam, and for the present we will assume that section as being the one we will use. We will call the approximate length of the cap-beams 18 feet. This would give a total weight for the cap-beams of 36 × 65 or 2340 pounds. Dividing this by 314.16 to distribute it over the tank area as before, we have 7.45 pounds per square foot. This amount is very small—about 6/10 of one per cent of 1212.5—so in practical work we could ignore it, as we would only be overloading the joists by that amount. As an example let us suppose that the

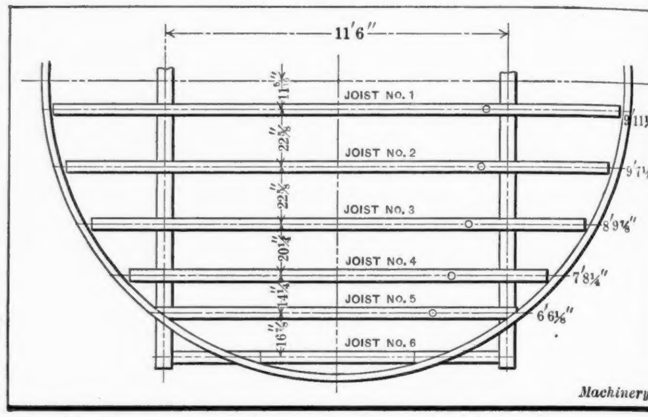

Fig. 15. Diagram used in determining Spacing of Joists in Floor System

fiber stress was 14,000 pounds per square inch when we omitted the weight of the cap-beams. When we include them the fiber stress will increase to about 14,000 × 1.006 = 14,084 pounds per square inch. A fiber stress of 14,000 pounds per square inch is used as an example because the increase would be larger in proportion than for a fiber stress of 12,500 pounds per square inch. To be theoretically correct we will omit the weight of the cap-beams in computing the joist loads, which will then become 1212.5 — 7.45 = 1205.05, which we will call 1205 pounds per square foot of tank bottom.

The maximum distance allowed between joists under wood tanks is 18 inches. This makes the maximum distance between centers of the joists 18 inches plus the width of the flange. We have to assume some size of beam in order to arrive at the approximate centers, and in case 10-inch, 25-pound beams are used, the maximum distance between centers would be 22.66 inches—say 22 5/8 inches. We next lay out a diagram, similar to that shown in Fig. 15, representing the outside of the tank by a circle 20 feet in diameter to some convenient scale. Then draw two parallel lines 11 feet 6 inches apart to represent the cap-beams, the 11 1/2 feet being obtained from the formula for finding the most economical centers for cap-beams. A line is then drawn representing the joist next to the center line of the tank, which will be 11 5/16 inches from it. Unless it is unavoidable, a beam should not be placed directly on the center line of the tank as that space is usually reserved for the riser pipe and its flanges. We will now determine the load on this joist which we will call joist No. 1. The "width of load" carried by this beam will be 22 5/8 inches. The "length of load" can be obtained quite accurately by scaling, if the diagram is drawn to a large scale, and is the distance across the tank (or chord) at the center line of the load width. However, we need only use the half chord, as we can obtain the bending moment of the joists just as well that way, and then obtain the concentrated loads on the cap-beam, due to the joists, directly. The load on one half the joist is equal to the product of the width of load multiplied by the length of load (one half chord) multiplied by the intensity of the load per square foot of tank bottom. That is:

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$$W = B\ L\ w$$

where W = load on half the joist = reaction on cap-beam due to joist;

B = width of load;

L = length of load or half chord;

w = load per square foot of tank bottom.

For joist No. 1 we have all the necessary data except the value of L. This can be found by solving a triangle having the tank radius as a hypothenuse, the distance of the center line of the load from the tank center as a perpendicular and the half chord as a base. This base is the unknown quantity L. For joist No. 1 we have 10 feet as a hypothenuse and 11 5/16 inches as the vertical. This gives us a length L of 9 feet 11 15/32 inches, which we will call 9 feet 11½ inches, working to thirty-seconds being a needless refinement as other results are not that accurate. We have now all the required factors and compute half the total load on joist No. 1 as follows:

$W = B\ L\ w = 1.885 \times 9.958 \times 1205 = 22{,}625$ pounds.

From this we can obtain the bending moment at the middle of the beam. We must then compute the overhang for the part of the joist extending beyond the cap-beam separately. Fig. 16 illustrates how the middle bending moment is obtained. The center of gravity of the load in a longitudinal direction is located midway of the distance L. For joist No. 1, $\frac{L}{2}$ is equal to 4 feet 11¾ inches or 59¾ inches. Then the distance X shown in Fig. 16 is:

$$X = 69 - 59\tfrac{3}{4} \text{ or } 9\tfrac{1}{4} \text{ inches;}$$

$$M = 22{,}625 \times 9\tfrac{1}{4} = 209{,}281 \text{ inch-pounds.}$$

where M is the bending moment at the middle of the joist.

We will now check this bending moment by another method. The maximum bending moment for a simple beam uniformly loaded is $\frac{W L}{8}$, where W is total load between supports located a distance L apart. When we make w equal the load per square foot and B the load width as before, the maximum bending moment becomes,

$$M = \frac{wBL^2}{8} \text{ foot-pounds} = 1.5\ wBL^2 \text{ inch-pounds.}$$

For an overhanging beam where the overhang in feet equals l we have the bending moment

$$M = \frac{wBl^2}{2} \text{ foot-pounds} = 6\ wBl^2 \text{ inch-pounds.}$$

From the foregoing, we obtain the following expression for the maximum bending moment for a beam on two supports with overhanging ends, similar to joist No. 1:

$$M = \frac{B\ w\ (L^2 - 4\ l^2)}{8} \text{ foot-pounds.}$$

$$M = 1.5\ B\ w\ (L^2 - 4\ l^2) \text{ inch-pounds.}$$

These two formulas apply only where the overhang is not more than 0.207 of the total length of the beam or, in this case, the chord of the tank at the point under consideration. In the case of a tank resting on beams, the length of the load is independent of the length of joist, so that the load could continue on beyond the end of the beam any distance—if it were possible—and the bending moment would be controlled by the length of the load.

Working out joist No. 1 by the last formula given, we have: $M = 1.5 \times 1.885 \times 1205 \times (132.25 - 4 \times 17.7101) = 209{,}277$ inch-pounds, which is the same as we obtained by the first method for all practical purposes. The first method is more accurate and simpler than the second as there are not as many figures to handle.

We have now found the bending moment at the middle of the beam but that is not necessarily the maximum moment. To determine whether this is the case we compute the bending moment due to the overhang. For joist No. 1 the overhang is 4 feet 2½ inches. The bending moment in inch-pounds, as previously stated, is:

$M = 6\ wBl^2 = 6 \times 1205 \times 1.885 \times 4.208^2 = 241{,}417$ inch-pounds. This is the maximum bending moment due to the fact that the overhang of the load is greater than 0.207 times the length of the joist between the cap-beams. This could have been ascertained in the first place, had we so desired, by dividing the overhang of 4 feet 2½ inches by the chord 19 feet 11 inches which would have given us 0.211, thus showing that the overhang created the maximum bending moment.

We have assumed a 10-inch, 25-pound I-beam for the joists. The section modulus for that size is 24.4 which gives a fiber stress of $\frac{241{,}417}{24.4} = 9894$ pounds per square inch. This value may seem low, but we will find that as the overhang grows shorter the maximum fiber stress in the joist increases, and that it becomes a maximum when no overhang exists. Joist No. 1 is the only one where the overhang is the controlling feature, although the maximum bending moment may not be found in the overhanging part of this joist, the location of the maximum moment depending upon the centers of cap-beams that were adopted.

We will next take up joist No. 2. By calculation, we find the length L to be 9 feet 7⅛ inches. The center of gravity of this load would be located at the middle of this distance or 4 feet 9 9/16 inches from center of tank. The distance X in Fig. 16 would then equal 69 — 57 9/16 = 11 7/16 inches. The amount of the load, assuming that it can still be 22⅝ inches wide, would then be:

$W = B\ L\ w = 1.885 \times 9.594 \times 1205 = 21{,}797$ pounds.

The middle bending moment is then found to be:

$M = 21{,}797 \times 11\ 7/16 = 249{,}303$ inch-pounds.

$249{,}303 \div 24.4 = 10{,}217$ pounds per square inch = fiber stress. The overhang equals 9 feet 7⅛ inches — 5 feet 9 inches = 3 feet 10⅛ inches. Dividing this overhang by the chord we have: $3.845 \div 19.188 = 0.20033$. As this is less than 0.207, the maximum bending moment occurs at the middle and will do so in the balance of the joists.

In figuring joist No. 3 we had better determine the width of load which it can support rather than figure the fiber stress from a given load width. However, we must assume the same width as before in order to arrive at the length L. Joist No. 2 can carry the maximum load width of 22⅝ inches, so we will place joist No. 3 at that distance from joist No. 2. Then for the length L we get 8 feet 9⅞ inches. We take the nearest ⅛ inch in this case. The bending moment equals $w\ B\ L\ X$, and solving for B we have:

$$B = \frac{M}{w\ L\ X};$$

$$M = 24.4 \times 12{,}500 = 305{,}000 \text{ inch-pounds;}$$

$$X = 69 - 52.9375 = 16.0625 \text{ inches.}$$

$$B = \frac{M}{w\ L\ X} = \frac{305{,}000}{1205 \times 8.8225 \times 16.0625} = 1 \text{ foot } 9\ 7/16 \text{ inches.}$$

This shows that joist No. 3 is not good for the maximum load width of 22⅝ inches. As the load over the space between two beams is distributed equally between them, we know that joists No. 2 and No. 3 each take 11 5/16 inches of the space of 22⅝ inches between them. Then if joist No. 3 is good for 21 7/16 inches, we can give it a load on the side between it and joist No. 4 of 21 7/16 — 11 5/16 = 10⅛ inches. This will make a center to center distance between joist No. 3 and No. 4 of 20¼ inches. Joist No. 4 is located 20¼ inches from joist No. 3 and we calculate its allowable load width in the same manner. The length L becomes 7 feet 8¼ inches. The distance X equals 69 — 46⅛ = 22⅞ inches.

$$B = \frac{M}{w\ L\ X} = \frac{305{,}000}{1205 \times 7.688 \times 22.875} = 1 \text{ foot } 5\tfrac{1}{4} \text{ inches.}$$

We have then an allowable load width of 17¼ inches for joist No. 4, and to get the distance to joist No. 5 we subtract 10⅛ from 17¼ which gives 7⅛ inches or 14¼ as the distance from joist No. 4 to joist No. 5. In calculating the half chord or length L for joist No. 5 we find it to be 6 feet 6⅛ inches. The distance X equals 69 — 39.06 = 29.94 inches.

$$B = \frac{M}{w\ L\ X} = \frac{305{,}000}{1205 \times 6.51 \times 29.94} = 1 \text{ foot } 3\ 9/16 \text{ inches.}$$

The distance to joist No. 6 will then be:

$$(15\ 9/16 - 7\tfrac{1}{8}) \times 2 = 16\tfrac{7}{8} \text{ inches.}$$

We have now located all the joints, as shown in Fig. 15, and find two faults with the spacing. The first is that joist No. 5 is so placed that it has a poor bearing on the cap-beam. The second fault lies in the fact that joist No. 6 is too far toward the outside of the tank, so that it cannot carry a proper share of the load, due to its cramped position. It is always necessary to place an auxiliary joist on top of it, as the staves of the tank should never rest on a joist. A good form of construction used for this purpose is shown in Fig. 17. It will be seen that the main joist is connected directly to the webs of the cap-beams, and the auxiliary or sub-joist is supported on it by means of two separators made of short pieces of beams or castings. The object of separating the main and auxiliary

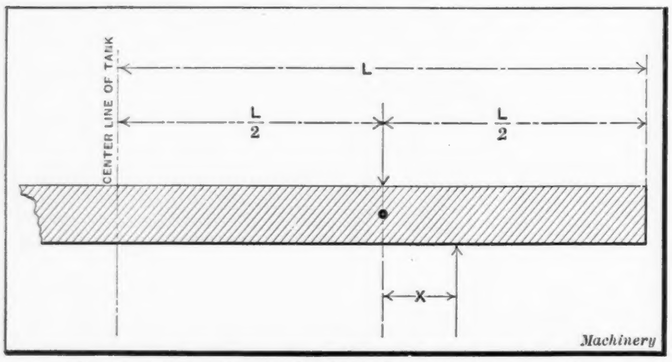

Fig. 16. Diagram illustrating Method of determining Bending Moments in Joists

joists in this way is to provide for securing the four-inch beams that are used to support the balcony. This construction will be taken up more fully later on.

We will now consider respacing the joists in order that they may all be placed in the most advantageous positions. It is necessary to move joist No. 5 a sufficient distance toward the center of the tank to give it a full bearing on the cap-beams and allow it to clear the tank staves by about an inch. One inch is the closest that the corner of any joist flange should come to the inside face of the tank staves; otherwise there may be trouble in erecting, as the tanks are not always of the exact dimensions called for. The variation is not large but it is not well to approach too close to the inside diameter. In order to locate joist No. 5 accurately, we will first figure the distance from the center line of the tank to the point where the inside circle of the staves crosses the line of the outside edge of the cap-beam flange as follows: The cap-beam centers are 11 feet 6 inches, and the flange width is 6¼ inches on a 20-inch, 65-pound I-beam. The thickness of the staves is about 3 inches. We now have a triangle to solve, having a hypothenuse of 9 feet 9 inches and a base of 6 feet ⅛ inch.

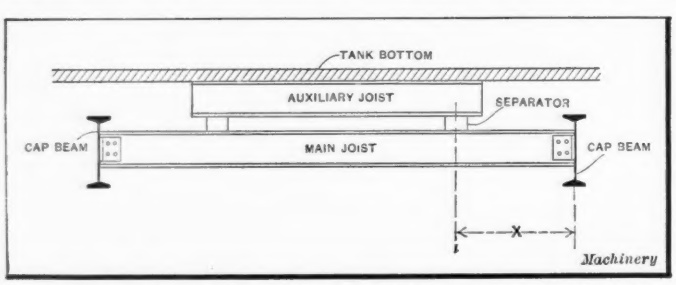

Fig. 17. Auxiliary Joist placed between Joist No. 6 and the Tank Bottom

The third side, which is the required distance of joist No. 5 from the tank center, is thus found to be 7 feet 8⅛ inches. The flange of a 10-inch, 25-pound I-beam is 4 11/16 inches wide. Half of that would be 2 11/32 inches, which we will call 2⅜ inches. Now if the outer edge of the flange of joist No. 5 just touched the point we have located, the center line of the beam would be 2⅜ inches nearer the center of the tank than the point referred to, or 7 feet 8⅛ inches — 2⅜ inches = 7 feet 5¾ inches. As previously mentioned, the flange should clear the inside face of the staves by about an inch and, therefore, we deduct one inch from 7 feet 5¾ inches which gives 7 feet 4¾ inches as the proper distance from the center line of joist No. 5 to center of tank. According to the spacing shown in Fig. 15, joist No. 5 is 91 1/16 inches from the tank center while we find it should not be more than 88¾ inches or 2 5/16 inches closer. We will also move joist No. 6 closer—say ⅞ inch—making it 16 inches from joist No. 5.

According to previous calculations, joist No. 5 was good for a load width of 16 inches. As we have 8 inches of load on one side (toward joist No. 6) we can have only 8 inches on the other side and therefore we must have 16 inches between joists No. 5 and No. 4. Joist No. 4 is good for 17¼ inches, and as it has 8 inches of load on one side there can be 17¼ — 8 = 9¼ inches on the other side. This gives a space of 18½ inches between joists No. 4 and No. 3. Joist No. 3 is good for 21 7/16 inches and if there are 9¼ inches on one side, there can be 21 7/16 — 9¼ = 12 3/16 inches on the other side. This would make a space of 24⅜ inches between joists No. 3 and No. 2, but as we have a limit of 22⅝ inches, we cannot exceed that figure although less would be preferable, as we should reduce the load if possible. Now we can respace the joists to best meet the requirement of existing conditions, and we will begin by eliminating the fractional inches in the spacing, still keeping the load limit we have worked out in mind. We will place joist No. 5 at a distance of 89 inches from the tank center, instead of 88¾, and space the remainder of the joists according to Fig. 18. From this the reader can see that the loads on the beams are apparently reduced to some extent. We must remember, however, that when we move a joist toward the center of the tank the length of the load increases, and as it will be necessary to go over the joists again to get the loads on the cap-beams, we can compute the fiber stress to see that they are not overloaded.

We have not yet done anything with joist No. 6, as it is necessary to decide on the final spacing in order to arrive at the correct dimensions of the segment over which the load that it carries is distributed. If the diagram is made to a

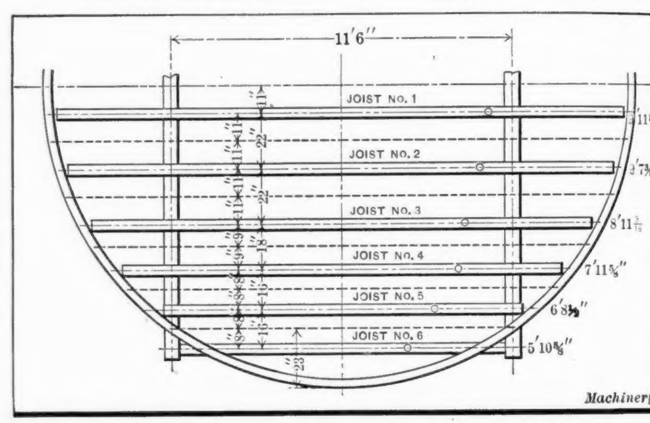

Fig. 18. Diagram showing Joists respaced to receive a Better Distribution of the Load

large scale, the dimensions can be scaled close enough for all practical purposes. We will, however, calculate all dimensions. Fig. 18 shows the division of the loads for each joist by means of dotted lines. Beginning with joist No. 1, we will proceed as before, using the same nomenclature, and determine the load and fiber stress for each joist with the arrangement shown in Fig. 18. As the reader is now familiar with the method used for this purpose, we will show the operations in their regular order and omit any explanation except where something new comes up.

Joist No. 1

$L = 9$ feet $11\frac{1}{2}$ inches;

$W = wBL = 1205 \times 1.83 \times 9.96 = 22{,}000$ pounds;

$X = 69 - 59\frac{3}{4} = 9\frac{1}{4}$ inches;

$M = WX = 22{,}000 \times 9.25 = 203{,}500$ inch-pounds.

$$\text{Fiber stress} = \frac{M}{Z} = \frac{203{,}500}{24.4} = 8340 \text{ pounds per square inch.}$$

This is the fiber stress for the bending moment at the middle of the beam which we have worked out in order to compare it with the bending moment of the overhang. The overhang is 9 feet 11½ inches — 5 feet 9 inches = 4 feet 2½ inches.

$M = 6\,wBl^2 = 6 \times 1205 \times 1.83 \times 4.21^2 = 234{,}100.$

$$\text{Fiber stress} = \frac{M}{Z} = \frac{234{,}100}{24.4} = 9621 \text{ pounds per square inch.}$$

The fiber stress due to the overhang is quite a little more than the stress for the middle bending moment.

Joist No. 2

$L = 9$ feet $7\frac{3}{8}$ inches;
$B = 22$ inches;
$W = w\ B\ L = 1205 \times 1.83 \times 9.62 = 21{,}240$ pounds;
$X = 69 - 57\ 11/16 = 11\ 5/16$ inches;
$M = W\ X = 21{,}240 \times 11.33 = 240{,}500$ inch-pounds.

$$\text{Fiber stress} = \frac{M}{Z} = \frac{240{,}500}{24.4} = 9848 \text{ pounds per square inch.}$$

Joist No. 3

The load on joist No. 3 is not balanced about the center line of the beam, as will be seen by referring to Fig. 18. It has 11 inches on one side and 9 inches on the other; and to find the length of load we will have to determine the distance of its center line from the center of the tank and proceed as before. The center of the load is $\frac{11 + 9}{2}$ or 10 inches from the side line or one inch off the center line of the joist. Then the distance of the center line of the load from the center line of the tank is $11 + 22 + 11 + 10 = 54$ inches or 4 feet 6 inches. Solving the triangle as before we have as the distance: $L = 8$ feet 11 3/16 inches.

$W = w\ B\ L = 1205 \times 1.66 \times 8.93 = 17{,}880$ pounds.
$X = 69 - 53\frac{5}{8} = 15\frac{3}{8}$ inches.
$M = W\ X = 17{,}880 \times 15.375 = 27{,}500$ inch-pounds.

$$\text{Fiber stress} = \frac{M}{Z} = \frac{27{,}500}{24.4} = 11{,}260 \text{ pounds per square inch.}$$

Joist No. 4

The center line of the load on joist No. 4 is one-half inch from the center line of the joist and nearer the center line of the tank. We calculate the distance in the same way as for joist No. 3.

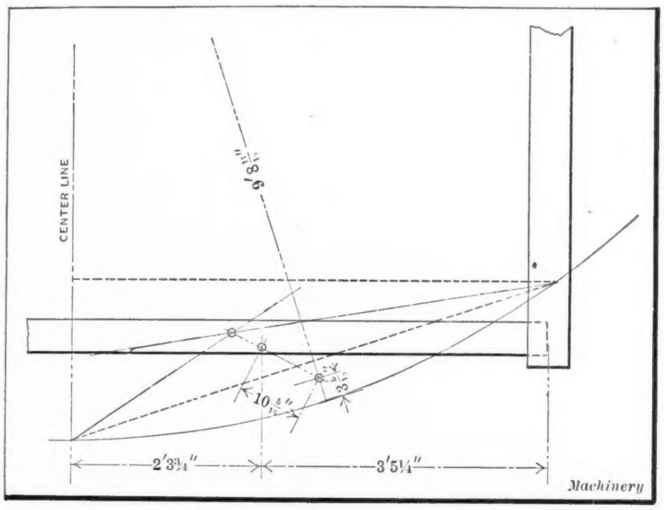

Fig. 19. Diagram used in determining Amount and Center of Gravity of Load on Joist No. 6

$L = 7$ feet $11\frac{5}{8}$ inches;
$W = w\ B\ L = 1205 \times 1.42 \times 7.97 = 13{,}603$ pounds;
$X = 69 - 47\ 13/16 = 21\ 3/16$ inches;
$M = W\ X = 13{,}603 \times 21.19 = 288{,}000$ inch-pounds;

$$\text{Fiber stress} = \frac{M}{Z} = \frac{288{,}000}{24.4} = 11{,}812 \text{ pounds per square inch.}$$

Joist No. 5

$L = 6$ feet $8\frac{1}{2}$ inches;
$B = 1$ foot 4 inches;
$W = w\ B\ L = 1205 \times 1.33 \times 6.71 = 10{,}778$ pounds;
$X = 69 - 40\frac{1}{4} = 28\frac{3}{4}$ inches;
$M = W\ X = 10{,}778 \times 28.75 = 309{,}900$ inch-pounds.

$$\text{Fiber stress} = \frac{M}{Z} = \frac{309{,}900}{24.4} = 12{,}700 \text{ pounds per square inch.}$$

This is slightly over the limit of 12,500 pounds per square inch but not enough to make it necessary to reduce the spacing. This we could do by moving joist No. 4 a little toward joist No. 5. The fiber stresses in joists No. 1 to No. 5 vary from 9621 to 12,700 pounds per square inch. We could reduce the size of the joists, if we placed a separator of some kind between them and the cap-beams to maintain the same level of their top surfaces. The question is, would it pay? The section modulus for a 9-inch, 21-pound I-beam is 18.9. If 9-inch beams were used, the fiber stress of joist No. 1 would be:

$$\frac{24.4}{18.9} \times 9621 = 12{,}420 \text{ pounds per square inch.}$$

The fiber stress for joist No. 2 would be:

$$\frac{24.4}{18.9} \times 9848 = 12{,}713 \text{ pounds per square inch.}$$

For joist No. 3 it becomes:

$$\frac{24.4}{18.9} \times 11{,}260 = 14{,}520 \text{ pounds per square inch.}$$

The stress in joist No. 3 with this sized beam is, of course, too large, but from the foregoing we can see that it would be possible to change the first two joists to 9-inch, 21-pound I-beams. We would thereby make a saving in weight of four pounds per lineal foot of joist thus altered. By the time we include the extra labor resulting from the change, it is doubtful if any saving would be made.

Joist No. 6

We will now take up the end joist No. 6. The first thing necessary is to compute the area of the segment over which the load it supports is distributed, as shown in Fig. 19. The area of a 20-foot circle is 314.16 square feet. The distance of the base of this segment from the tank center is 97 inches. This leaves 23 inches as the height of the segment, as shown in Fig. 18. Next we obtain half the chord in the same way as for the other joists, and it is found to be 5 feet $10\frac{5}{8}$ inches. The included angle between the extremes of the total chord of 11 feet $9\frac{1}{4}$ inches is found by first dividing half the chord by the tank radius. This gives the sine of half the included angle, from which the desired result can be obtained. Thus:

$$\sin\frac{a}{2} = \frac{5.885}{10.0} = 0.5885;$$
$$a = 72 \text{ degrees } 6 \text{ minutes.}$$

Then we get the area of the sector included in this angle from the following proportion:

$$\text{Area} = \frac{72 \text{ deg. } 6 \text{ min.}}{360 \text{ deg.}} \times 314.16 = 62.92 \text{ square feet.}$$

We next find the area of the triangle which has the chord for its base and the perpendicular distance 8 feet 1 inch from the center of the tank to the segment as its altitude. This area is:

$$8.083 \times 5.885 = 47.58 \text{ square feet.}$$

Then $62.92 - 47.58 = 15.34$ square feet, area of segment over which the load on joist No. 6 is distributed. All we need to consider is one-half the load, according to the method used on the other joists.

$15.34 \div 2 = 7.67$ square feet = area of half segment.
$7.67 \times 1205 = 9242$ pounds = load on half segment.

The next step is to find the center of gravity of the half segment in order that we may arrive at the bending moment of the joist. We divide the area into two parts, a triangle and a segment, as shown in Fig. 19. The center of gravity of the triangle can be found by bisecting the two adjacent sides and drawing diagonal lines as shown. The point where these diagonals intersect is the center of gravity of the triangle. Next we must find the center of gravity of the small segment. According to Kent and Trautwine, the distance of the center of gravity of a segment from the center of the circle of which it is a part (the tank in this case) is equal to:

$$\frac{\text{cube of chord}}{12 \times \text{area of segment}}.$$

We have the area for the total half segment, which equals 7.67 square feet. To get the area of the small segment, now under consideration, we must deduct the area of the triangular portion of which we just found the center of gravity from the total half segment. The area of the triangle is:

$$\frac{5.885 \times 1.916}{2} = 5.64 \text{ square feet}$$

Then 7.67 — 5.64 = 2.03 square feet = area of small segment. In order to use the formula given, we also need the length of the chord for the small segment. This we can obtain by working out the hypothenuse of the triangular portion which is found to be 6 feet 2¼ inches. Then the distance of the center of gravity of the segment from the center of the circle is:

$$\frac{\text{cube of chord}}{12 \times \text{area of segment}} = \frac{6.18^3}{12 \times 2.03} = 9 \text{ feet } 8\ 11/16 \text{ inches;}$$

10 feet — 9 feet 8 11/16 inches = 3 5/16 inches = distance from the outside of the tank to the center of gravity of the segment. Now we draw a line joining the centers of gravity of the triangle and the segment. This line is 14 inches in length and on it we must find the location of the center of gravity of the whole area. This we can do by the following proportion:

$$(5.64 + 2.03) : 5.64 = 14 : X.$$

where X is the distance from the center of gravity of the small segment to the center of gravity of the whole area under con-

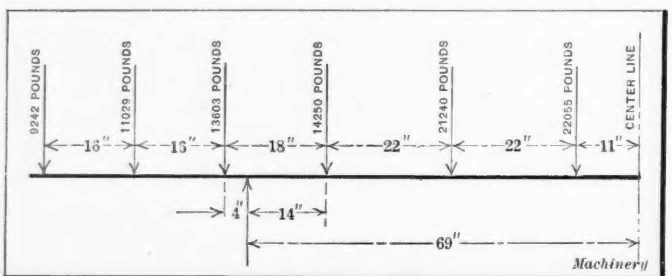

Fig. 20. Amount and Distribution of Loads applied on Cap-beams by Joists

sideration. From this proportion we find X = 10.295 inches. We will call it 10 5/16 inches. This last point is located 2 feet 3¾ inches from the center of the tank and 3 feet 5¼ inches from the center line of the cap-beam, as shown in Fig. 19. This is the point of application of the load 9242 pounds on joist No. 6.

Referring again to Fig. 17 we see that the separator is located a distance X from the center of the cap-beam. We must now decide where that separator can be located in order to use a 10-inch I-beam for the main joist, if possible. The load is 9242 pounds, the section modulus 24.4, and the allowed fiber stress 12,500 pounds per square inch. Then the distance

$$X \text{ equals } \frac{12{,}500 \times 24.4}{9242} = 33 \text{ inches from the center of the cap-}$$

beam or 36 inches from the middle of the joist. The longest length we can use for the auxiliary joist shown in Fig. 17 is 7 feet 7 inches. This places the separator 3 feet 9½ inches — 3 feet = 9½ inches from the end of the auxiliary joist. As the load is applied 27¾ inches from the center line of the tank, as shown in Fig. 19, the distance X in Fig. 16 becomes 36 — 27¾ = 8¼ inches on the sub-joist. The middle bending moment is then 76,246 inch-pounds. With 12,500 pounds as the fiber stress we need a section modulus of 6.1. The bending moment due to the overhang is 34,580 inch-pounds. For the sub-joist we can then use a 6-inch, 12¼-pound I-beam. As a rule the joists are all made the same size.

Cap-beams

The next thing in line is to determine the size required for the cap-beams. We have the loads on all the joists, but in figuring the bending moment we must include the weight of the beams themselves. This can be done in two ways. By increasing the loads due to the joists an amount that would take care of the weight of the cap-beams, or by leaving the joist loads as they are and calculating separately the bending moment in the cap-beams due to their weight and adding it to the bending moment due to the joists. The latter method is the easier one and will be used in the present discussion. Fig. 20 gives the arrangement of the loads applied on the cap-beams by the joists. There are two ways of obtaining the bending moment due to the joists. One is to add together the moments of all the loads about the center line and subtract their sum from the moment of the reaction about the center line. This gives the middle moment; then to get the end moment due to the overhang, we take the sum of the moments of the loads on the overhanging portion about the column center.

The second method, which is a shorter one, is to take all moments about the column center. For the middle moment take the inside loads, and for the end moment take the overhanging loads. The maximum moment is the difference between the first two moments, as follows:

14,250 × 14 = 199,500 inch-pounds.
21,240 × 36 = 764,640 inch-pounds.
22,055 × 58 = 1,279,190 inch-pounds.

Total 2,243,330 inch-pounds, middle moment.

13,603 × 4 = 54,412 inch-pounds.
11,029 × 20 = 220,580 inch-pounds.
9242 × 36 = 332,712 inch-pounds.

Total 607,704 inch-pounds, end moment.

From this we can see the overhang is not the controlling factor and we obtain the maximum moment due to the joists as follows: 2,243,330 inch-pounds — 607,704 inch-pounds = 1,635,626 inch-pounds, maximum moment.

We assumed to start with, that a 20-inch, 65-pound I-beam was large enough, but to get closer to the right size we will use the moment due to the joists to arrive at the proper size and change it later, if necessary.

$$\frac{1{,}635{,}626}{13{,}000} = 125.8 = \text{section modulus.}$$

This calls for a 20-inch, 75-pound I-beam. The approximate length of the cap-beams is 18 feet. We will only consider half of the beam. Then:

9 × 75 = 675 pounds, weight of half the cap-beam;

X = 69 — 54 = 15 inches (Fig. 16);

M = 675 × 16 = 10,125 inch-pounds, middle moment in cap-beam, due to its own weight.

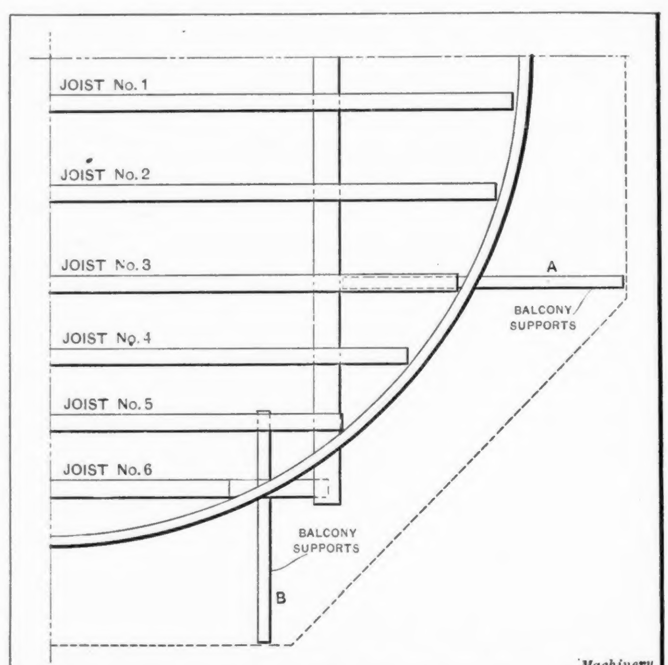

Fig. 21. Quarter Plan of Floor System, showing Beams for supporting Balcony

For the end moment we have:

3.25 × 75 × 19.5 = 4753 inch-pounds;

10,125 — 4753 = 5372 inch-pounds, maximum moment in cap-beam due to its own weight.

1,635,626 + 5372 = 1,640,998 inch-pounds, maximum bending moment in cap-beam.

$$\frac{1{,}640{,}998}{117} = 14{,}026 \text{ pounds per square inch, fiber stress in}$$

a 20-inch, 65-pound I-beam.

If we had used the weight of a 65-pound beam instead of a 75-pound beam, the fiber stress for the 65-pound beam would be reduced by about 5½ pounds per square inch and we would have a fiber stress of 14,020 pounds per square inch for the 20-inch, 65-pound I-beam. Now the designer can take his choice of the two sizes. It would not be out of the way to use

the lighter beam, according to present practice, although we would only have a factor of safety of about 2¼. There is one more factor to consider in the selection of beams to carry given loads, and that is their shearing strength. While a beam may be strong enough to carry the load when applied as a bending moment, it may be weak in its ability to resist shearing stresses. Therefore, a shallow beam should be selected in preference to a high one. For instance, a 20-inch, 80-pound I-beam is sometimes preferable to a 24-inch, 80-pound I-beam for the reason just mentioned. In the case under consideration, a 20-inch, 65-pound I-beam is of ample size to carry the load in shear.

We have now proportioned the joists and cap-beams, and the only point that remains is to determine the size of the beams that support the balcony. The balcony should be made two feet wide inside the hand rail, and that determines the size of the beams. They are located as shown in the quarter plan of the floor system, Fig. 21. It is customary to use 4-inch, 7½ pound I-beams, and with 3½ feet overhang that size would support 900 pounds at the end with a fiber stress of 12,600 pounds per square inch, which load is probably more than the beam will ever have to carry. Beam *A* is hung under joist No. 3 by riveting, or any other secure method. Beam *B* is placed on top of main joist No. 6 with its end under joist No. 5, and is bolted in place. The balcony is made by placing wooden beams of the proper size across the balcony beams, the planking being nailed to these stringers.

If a steel tank is to be used in place of a wood tank, the calculations would be made in the same manner, except that the beams used as joists would have to be spaced to suit the thickness of the bottom, according to the table entitled "Maximum Centers of Beams for Steel Tank Bottoms" printed in the July, 1911 issue of MACHINERY. The formula for the most economical centers of cap-beams and columns would then be changed to read: Column and cap-beams centers = 0.586 *D*, where *D* is the outside diameter of the tank.

In a wood tank, the sides are slanting inward toward the top and this throws the load slightly toward the center. Another thing which might be mentioned is, that the maximum distance of 18 inches between the flanges of the joists previously given, is for 3-inch lumber in the tank bottom. For lumber 2 inches and less in thickness, the distance should be lessened in proportion to the strength, which would be gaged by the section moduli of equal widths compared.

* * *

The position of the fabricators of steel, especially the manufacturers of drop-forgings, has changed greatly during the past ten years with respect to the specifying of steel. Formerly it was the custom for the purchaser of drop-forgings to leave the selection of the steel to the drop-forge concern. The makers of drop-forgings, of course, selected the low carbon steels which are easily worked. Now the customer, especially the large ones, specifies minutely the chemical and physical characteristics of the steel to be used, and in some cases even the name of the steel company from which the steel shall be bought. The drop-forge manufacturer is relieved of responsibility for the steel used, but has greater difficulties imposed on him by the new conditions. Not only are the steels harder to work, but they must be put through special heat-treatments, the details of which also are minutely prescribed.

* * *

The British machine tool builders' association, known as the Machine Tool & Engineering Association, Ltd., has issued a very optimistic report for the past year. This report points out that the machine tool exhibition at Olympia was very successful, both from the point of view of the association as a whole and from that of the individual members who exhibited. The total number of exhibitors was 292 and the total attendance about 100,000. The association was recently invited to organize a collective exhibit of machine tools at the Ghent exhibition this year, but the response to the invitations from the individual members was so inadequate that the scheme has been abandoned. The financial condition of the society is very satisfactory, the balance on hand after meeting all current liabilities amounting to nearly £4000 ($20,000). During the past year twenty-three members have joined the association, bringing the membership up to slightly over one hundred.

MILLING MOTOR TRUCK SPRING CHAIRS

BY GEORGE E. POPE*

In determining the necessary tools for machining a spring chair of the design shown in Fig. 1, the problem of finishing the surface *A* came up. As this surface has a radius of 19⅞ inches it was, of course, out of the question to use a cutter of this radius, even if such a method were possible. It was appreciated at the outset that the logical method of finishing this surface was to make a large fixture for the vertical boring mill, set the spring chairs around on the proper radius, and thus finish a number of them at one operation. This idea was discarded because a fixture of this sort represented a considerable first cost and there was only a small number of the spring chairs to be made.

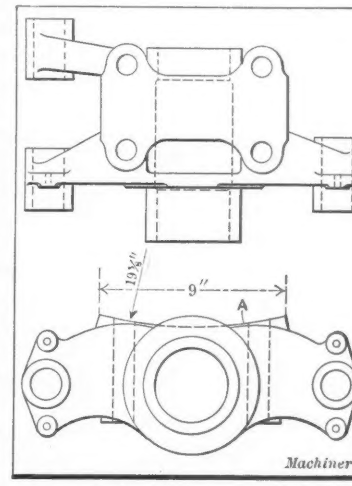

Fig. 1. Spring Chair for which Method of finishing Surface A was devised

Then a scheme was hit upon by which the surface *A* could be milled by the use of a fixture of comparatively small cost. This scheme, as illustrated in Fig. 2, was to design a suitable fixture for the horizontal milling machine; this fixture located the chair in a horizontal position from the large hole, which was previously finished to size. The largest cutter in stock was used in connection with a large Brown & Sharpe vertical milling attachment of the universal type. The cutter was set at an angle and the cutting was done by its edge as shown. It is evident that this method did not finish the surface with a true radius but left it elliptical. This was permissible, however, as these surfaces merely form seats for the truck springs and they are not expected to fit exactly, even if this surface is finished to a true radius.

In selecting a cutter, it can be seen, first, that the larger the cutter used the nearer the curve will approximate a circle; and second, that the diameter of the smallest cutter that can be used must be greater than the length of the surface to be milled. In fact, it will be readily seen that the smallest permissible diameter, greater than the length of the surface, will be determined by the limit of variation from a true circle that can be allowed. In determining the angle at which to set the cutter, the first consideration, after the diameter has been selected, is to find the minor axis of the ellipse. From the diagram shown in Fig. 3, a formula can be derived that will give the minor axis and also one that will give the amount of deviation from a true circle for any distance out from the center. For the derivation of a formula to determine the minor axis, the equations for a circle and an ellipse are employed, using rectangular coordinates with origins at O and O_1, respectively. These equations are $x^2 + y^2 = r^2$ for the circle and $\frac{x^2}{a^2} + \frac{y^2}{b^2} = 1$ for the ellipse. The notation in Fig. 3, which shows an ellipse superimposed upon a circle with the vertical center lines coincident and with the perimeter of the ellipse tangent to the circumference of the circle, is as follows: x = the distance from the common center line to the point where it is desired that the circle and the ellipse will coincide and is the common abscissa of both the ellipse and the circle. b = one half the minor axis of the ellipse. y_1 = the ordinate of the point on the ellipse. a = one-half the major axis of the ellipse. r = the radius of the circle it is desired to approximate. y = the ordinate of the point on the circle. Then with y_1 and b as the unknowns, simultaneous equations must be found that will eliminate y_1, as b is the dimension required. In other words, what is desired is the minor axis of the only ellipse that can be drawn through the point on the circle of which x is the abscissa, when r and a are given, and with center

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lines coincident and the outlines tangent. Then from the equation for the ellipse is derived

$$y_1 = \frac{b}{a}\sqrt{a^2 - x^2}$$

From Fig. 3 is obtained $r - b = y - y_1$, which gives $y_1 = b + y - r$. Therefore

$$b + y - r = \frac{b}{a}\sqrt{a^2 - x^2}$$

from which

$$b = \frac{a\,(y - r)}{\sqrt{a^2 - x^2} - a}$$

From equation $x^2 + y^2 = r^2$ is derived $y = \sqrt{r^2 - x^2}$

and substituting $b = a\,\dfrac{\sqrt{r^2 - x^2} - r}{\sqrt{a^2 - x^2} - a}$

For the case of the spring chairs: $r = 19\frac{5}{8}$ inches, the chordal length of the surface to be milled = 9 inches, making $x = 4\frac{1}{2}$ inches; the cutter used is 14 inches in diameter, making $a = 7$ inches. Substituting these numerical values

$$b = 7\,\frac{\sqrt{19.625^2 - 4.5^2} - 19.625}{\sqrt{7^2 - 4.5^2} - 7} = 2.2341 \text{ inches.}$$

Now turning to Fig. 4 the angle at which the milling head must be set can be obtained. It can be seen by this diagram

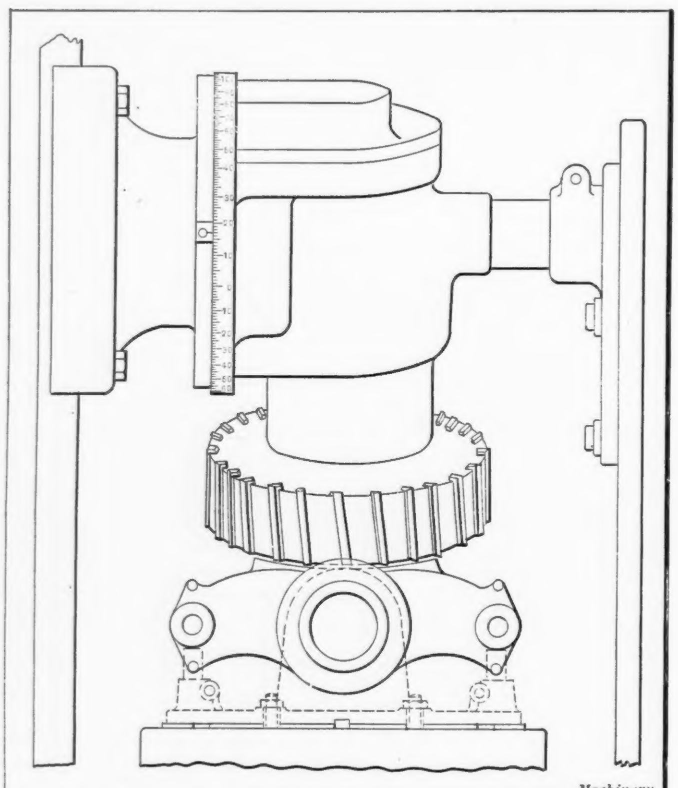


Fig. 2. Method of finishing Surface A on the Milling Machine

that if $2b$ represents the minor axis of the ellipse, and $2a$ the major axis of the ellipse and also the diameter of the cutter, $b \div a$ will equal the sine of the angle desired. This, for the example at hand, equals $2.2341 \div 7 = 0.3191$ for the sine. This gives an angle of 18 degrees 37 minutes.

It can be readily seen that with the respective center lines coincident it is impossible for an ellipse and a circle to coincide except at one point in each quadrant of the ellipse. Returning to Fig. 3 it is evident then that there is a deviation of the outline of the ellipse from the circumference of the circle along the length of x. The respective outlines are, of course, tangent at the center line. This deviation can be investigated at different distances out from the center line in order to decide, by finding its maximum amount, whether this method of approximating a circle is accurate enough for the purpose under consideration. It must be remembered, as stated in a preceding paragraph, that this deviation can be diminished to any desired amount, provided the diameter of the cutter can be sufficiently increased. In other words, the deviation becomes zero when the radius of the cutter equals the radius of the surface to be milled. This investigation can be conducted by holding a and b constant, selecting different values for x, and then finding y_1 in the formula for an ellipse. From the formula for the ellipse the value of y_1 was obtained.

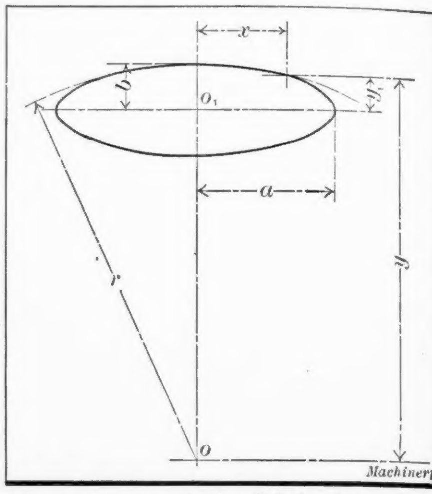


Fig. 3. Diagram showing Relation between the Desired Circle and the Ellipse actually obtained

Then in the equation for the circle there will, be, of course, a new value of y for every different value of x that is selected. This value can be found by using the equation $y = \sqrt{r^2 - x^2}$. Selecting distances from the center line of 1, 2, 3 and 4 inches, y and y_1 equal the values tabulated below:

	1	2	3	4
y	19.5995	19.5228	19.3943	19.2130
y_1	2.2108	2.1406	2.0182	1.8329

Consulting the diagram in Fig. 3 again, it will be seen that $(r - b + y_1) - y$ will be an expression by which the deviation of the outline of the ellipse from the circumference of the circle can be found after knowing the various values of y and y_1. It is found to be 0.0022 inch, 0.0087 inch, 0.0148 inch and 0.0108 inch, respectively, at the points where x has values of 1, 2, 3 and 4 inches.

It will be noticed from these results that the outline of the ellipse rises quite rapidly away from the circumference of the circle, after crossing it at the point 4.5 inches from the center, and then gradually declines toward it again as it proceeds toward their common point of tangency. As 0.0148 inches is probably very nearly the maximum of deviation of the ellipse from the circle, this proved to be a very satisfactory way of milling these spring chairs.

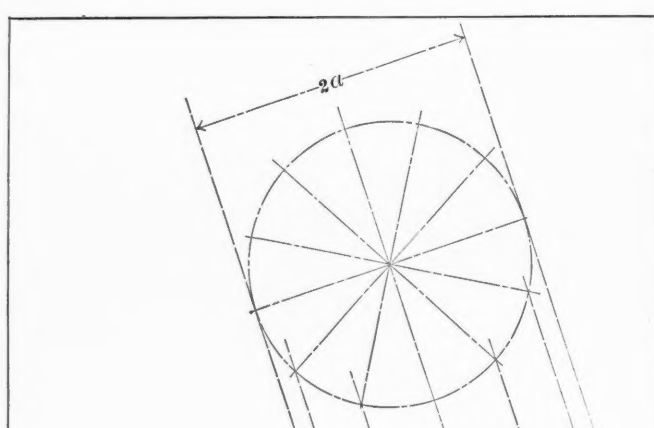


Fig. 4. Diagram used to determine Angle at which to set the Milling Cutter

* * *

Two French aviators, Lieuts. Varcin and Battini, have in six months covered 12,000 miles and carried 600 passengers, without any accident.

CROSS-DRILLING AND MILLING ATTACHMENTS*

APPLICATION TO THE "ACME" MULTIPLE-SPINDLE AUTOMATIC SCREW MACHINE

BY DOUGLAS T. HAMILTON†

Many parts which ordinarily require a second operation can be completed in one on the "Acme" multiple-spindle automatic screw machine by the use of special attachments. These are easily and quickly put into position, and owing to the "Acme" method of tooling, do not retard the output, except when extra heavy cuts are the longest single operation. In that case the time for the additional operation is the time required to make one piece, the regular operations being completed at or during the same time. The "Acme" automatic screw machine lends itself admirably to the application of special attachments, because of the stopping of the spindle in the third position for threading. This saves the cost of rehandling, and in the great majority of cases does not take any extra time.

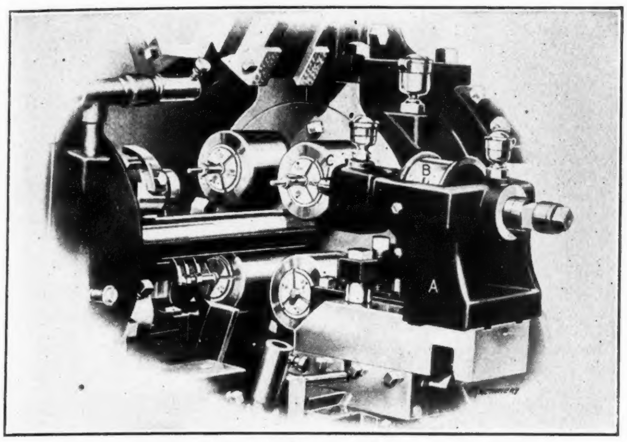


Fig. 1. Cross-drilling Attachment held on Cut-off Tool-slide

Cross-drilling Attachment held on Side-working Tool-slide

One of the standard cross-drilling attachments used on the "Acme" machine is shown in Fig. 1. This consists of a cast-iron frame *A* which is bolted to the top face of the cut-off tool-slide and works in the third position, where the work-spindle can be stopped. The cross-drilling and threading operations can usually be performed at the same time. The drive for this attachment is by a flat belt from a special overhead countershaft running on the pulley *B* which is fastened to the spindle *C* that carries the drill. This attachment, by a slight modification in its construction, can be driven by gears and a universal-joint shaft from the main tool-slide.

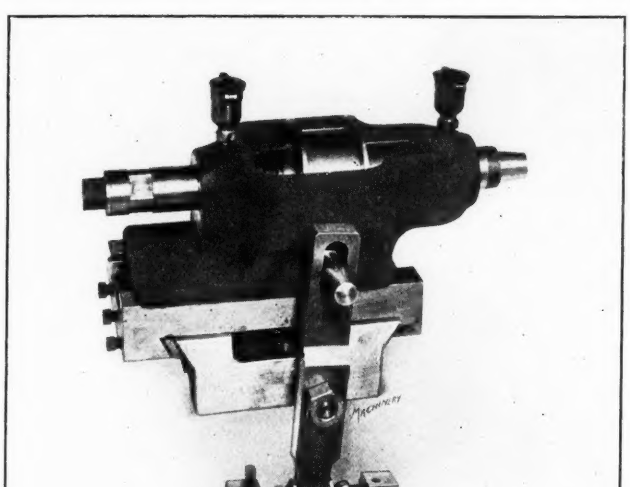


Fig. 3. Cross-drilling Attachment shown in Fig. 2 dismantled from the Machine

Cross-drilling Attachment with Accelerating Movement

Another cross-drilling attachment, held in a similar position to that just described but provided with an accelerating movement for increasing the travel of the drill, is shown in Figs. 2 and 3. Fig. 2 shows this attachment set up on the machine, and Fig. 3 shows it removed from the cut-off tool-slide. The attachment *A* is similar in construction to that shown in Fig. 1, except that it is mounted on two slides *B* and *C*. Slide *C* is fastened to the top face of the cut-off tool-slide, and slide *B* fits over the former and is furnished with a gib to provide for adjustment. This enables the drilling attachment to be moved longitudinally along the base, facilitating adjustments for the drilling of holes at different distances from the face of the chuck. Attachment *A* is operated by a lever *D* which is fulcrumed to the lower slide *C*.

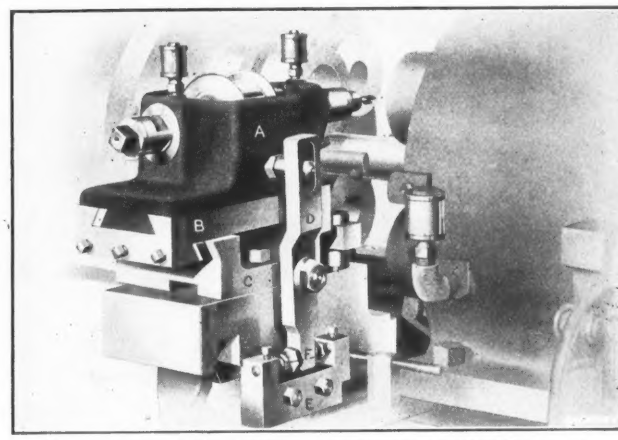


Fig. 2. Cross-drilling Attachment with Accelerating Movement

A block *E* provided with hardened adjustable stops *F* is fastened to the base in which the cut-off tool-slide works. This block, by means of its adjustable points, stops the lower portion of lever *D*, so that instead of following the movement of the cut-off tool-slide when it is fed in, it transmits a movement to the lower arm of the lever and thus accelerates the travel of the drill-holder. The ratio between the arms of lever *D* is 1¾ to 1, thus making it possible to drill a hole clear through a piece. The regular travel of the cross-slide is only equal to a little over one-half the diameter of the bar, so that when it is necessary to drill a hole entirely through the work, this attachment with accelerated movement can be used to very good advantage.

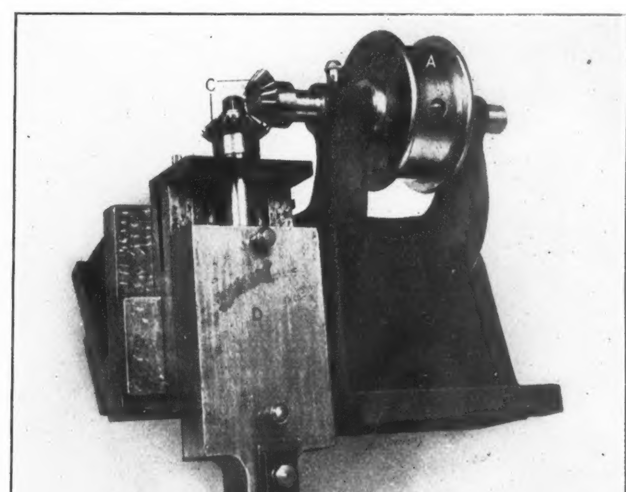


Fig. 4. Cross-drilling Attachment operated from the Top-working Tool-slide in the Third Position

* For further information on National-Acme automatic screw machine equipment, see "Automatic Screw Machine Equipment," March, 1913, and February, 1913, and articles there referred to.

† Associate Editor of MACHINERY.

Cross-drilling Attachment held on Top-working Tool-slide

A cross-drilling attachment, mounted on the regular top bracket, or a special bracket taking its place, and operated in the second position, is shown in Fig. 4. This attachment is driven from a special countershaft. The slide *D* in which the drill spindle *B* is located is adjustable sideways, thus enabling holes to be drilled in different locations in relation to the face of the chuck. This attachment is ordinarily used in conjunction with other attachments operating from the side or end, when the operations must be performed at angles to each other. The bracket shown in the illustration is used for drilling holes at right angles, but holes at practically any

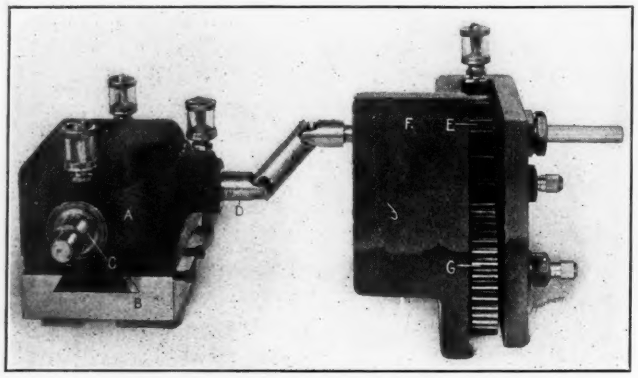


Fig. 5. Cross-drilling Attachment driven by Gears and a Knuckle-joint Shaft from the Threading Spindle Gears

angle can be drilled by the substitution of a bracket suitable for the case in hand.

Cross-Drilling Attachment held on Side-working Tool-slide and operated from the Main Tool-slide

A cross-drilling attachment which is held on the top face of the cut-off tool-slide and operated from the main tool-slide by the gears used in driving the threading spindle, is shown in Fig. 5. This attachment is particularly adapted for use on the motor-driven machines, but may be used equally well on the belt-driven machines, where it will eliminate the special countershaft. The top casting *A* is gibbed in the base *B*,

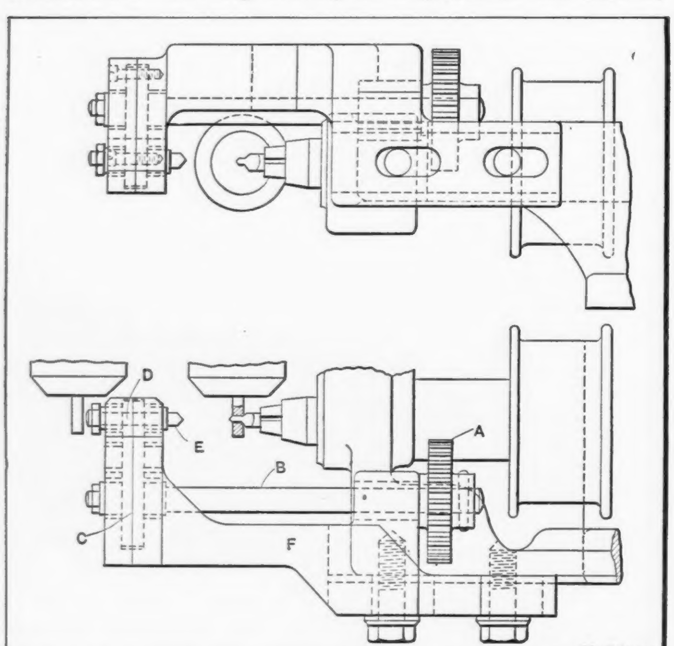


Fig. 6. Cross-drilling Attachment with Opposite Spindles

which is fastened to the top face of the cut-off tool-slide and travels with it. The drill spindle *C* is driven by bevel gears receiving motion from the knuckle-joint shaft *D*, which is splined to fit a key in the gear *E*. The bracket *F* is bolted to the main tool-slide, and the gear *G* meshes with the gear driving the threading spindle.

Cross-drilling Attachment with Opposite Spindles

The cross-drilling attachment shown in Fig. 6 is provided with opposite spindles and is adapted for drilling cross holes, and, in addition, counterboring or countersinking from both sides. It can also be used for drilling parallel holes of the same or different diameters at a given distance from each other and from the face of the chuck. The holes can either be drilled entirely through the work or to any distance desired. When necessary, the attachment can be provided with

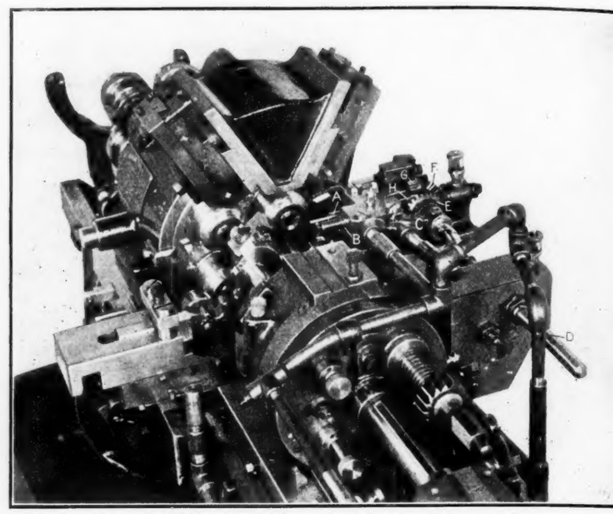


Fig. 7. Cross-drilling and Milling Attachment driven by Gears and a Knuckle-joint Shaft from the Threading Spindle Gears

an accelerating device for increasing its travel. The second or auxiliary spindle of this attachment is driven by spur or bevel gears from the regular drill spindle. When driven by spur gears, the drive is through gear *A*, shaft *B* and gears *C* and *D*. Gear *D* is keyed to the spindle in which the counter-

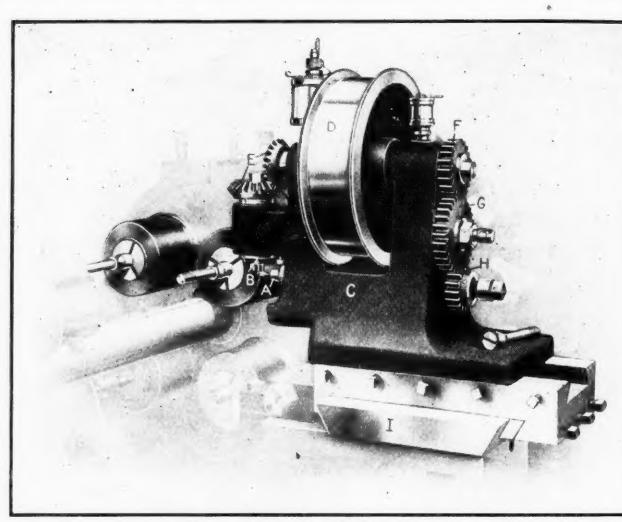


Fig. 8. Cross-drilling and Milling Attachment with Spindles located at Right Angles

sink *E* (or drill) is held. The bracket *F* carrying the auxiliary mechanism is bolted to the front side of the regular attachment used for cross-drilling. The type of attachment which this auxiliary device is used in connection with is shown in Fig. 1.

In operation, as the cylinder indexes, the stock comes be-

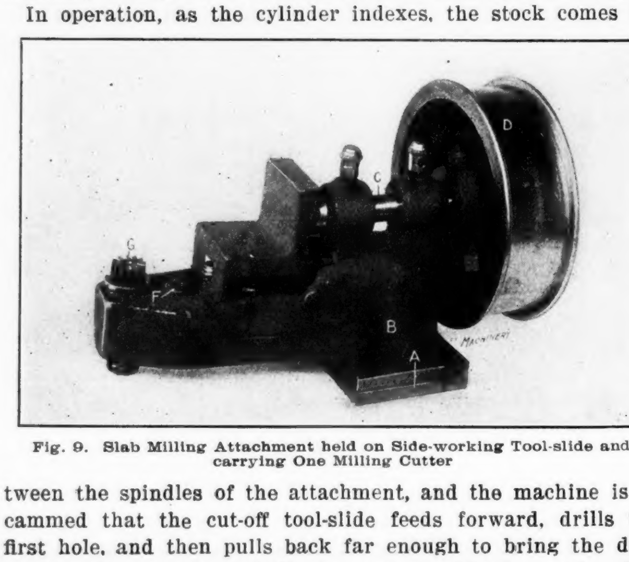


Fig. 9. Slab Milling Attachment held on Side-working Tool-slide and carrying One Milling Cutter

tween the spindles of the attachment, and the machine is so cammed that the cut-off tool-slide feeds forward, drills the first hole, and then pulls back far enough to bring the drill held in the opposite spindle in contact with the work. The

slide then feeds forward part way again before the next indexing operation.

Combination Cross-drilling and Milling Attachments

The attachment shown in Fig. 7 is provided with two horizontal spindles *A* and *B* holding a drill and an end milling cutter, respectively. The frame of this attachment is fastened to the top face of the cut-off tool-slide, and the tools are oper-

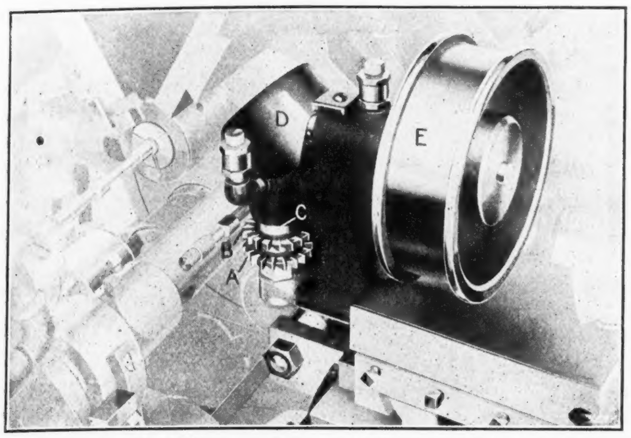


Fig. 10. Slab Milling Attachment carrying Two Milling Cutters

ated in the third position. The spindles *A* and *B* are driven from the knuckle-joint shaft *D* by gears meshing with the gears driving the threading spindle, as is shown more clearly in Fig. 5. Power is transmitted by this knuckle-joint shaft to the drilling and milling spindles by means of the bevel

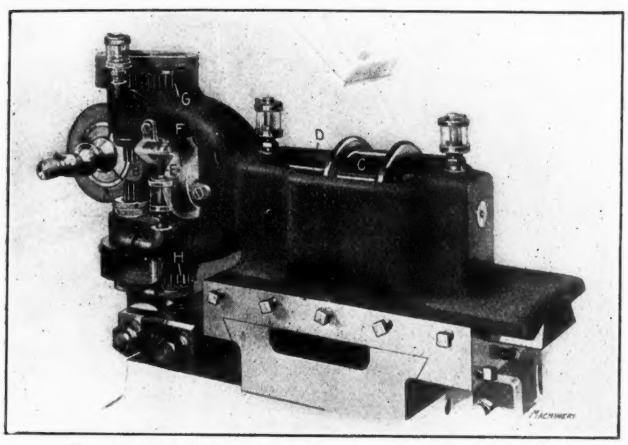


Fig. 11. Slab Milling Attachment carrying Two End-mills

gears *E* and *F* and spur gears *G* and *H*. Spur gear *H* forms an integral part of the drill spindle, which also carries the bevel gear *F*. This attachment is brought into position and fed toward the work by the arm operating the cut-off tool-slide.

A two-spindle drilling and milling attachment in which the

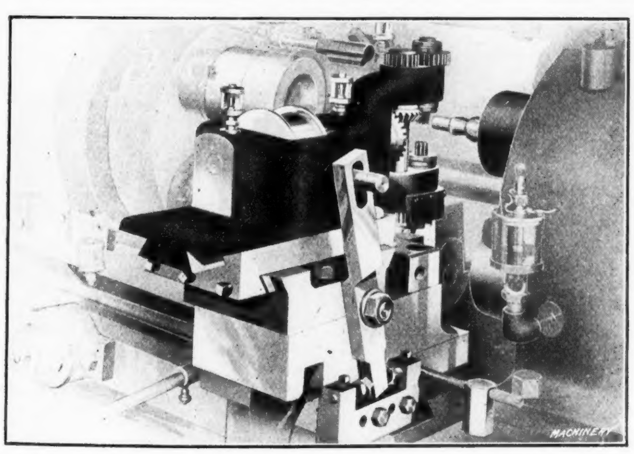


Fig. 12. Attachment shown in Fig. 11 provided with Accelerating Movement

spindles are located at right angles to each other is shown in Fig. 8. This attachment is used for drilling a cross hole and milling a flat on the work as indicated by the piece lying on the base of the attachment. The casting *C* which carries the spindles *A* and *B* is fastened to the top face of the cut-off tool-slide, and carries a driving pulley *D* which is driven through a flat belt from a special overhead countershaft. Pulley *D* is keyed to the top horizontal shaft and drives the vertical milling spindle through bevel gears *E*. On the rear end of the top horizontal shaft is a spur gear *F* which, through the intermediate gear *G*, drives the spur gear *H* fastened to the drilling spindle *A*. This attachment is adjustable longitudinally on the base *I*, the latter being fastened to the top face of the cut-off tool-slide. The attachment can be provided with an accelerating movement if desired.

A milling attachment, held on the side-working tool-slide and carrying one cutter or end-mill presented in a vertical position, is shown in Fig. 9. The base *A* of this attachment

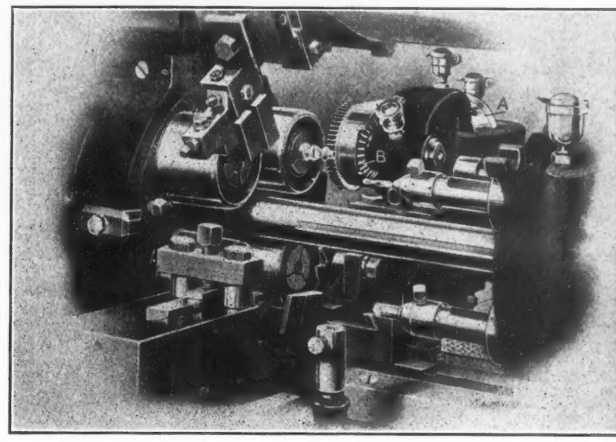


Fig. 13. Slab Milling Attachment driven by Bevel Gears

is fastened to the top face of the cut-off tool-slide, and carries a bracket *B* provided with bearings for holding the horizontal spindle *C*, which is driven from the overhead countershaft by a belt running on pulley *D*. Spindle *C* drives another horizontal spindle (enclosed in the fixture), through bevel gears which are covered by the guard *E*. The horizontal shaft inside the fixture drives the vertical stud *F* through bevel gears. Stud *F* carries a spur gear meshing with another spur gear on the cutter-spindle *G*. This attachment is used in the third position, the work-spindle being locked when milling is under way.

Vertical Spindle Milling Attachment carrying Two Cutters

Fig. 10 illustrates a vertical-spindle slab milling attachment, designed for carrying two face milling cutters *A* and *B*.

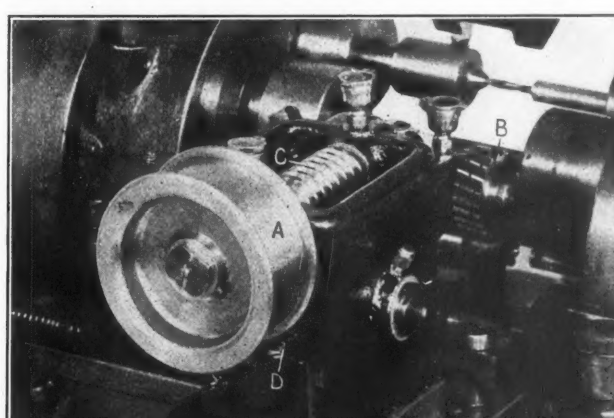


Fig. 14. Slab Milling Attachment driven by a Worm and Worm-wheel

These cutters are held on the vertical spindle *C* and are separated by a spacing washer of the required thickness. The attachment is held on the top face of the cut-off tool-slide, and is arranged for milling two flats on a cold-rolled steel piece, which is turned out at the rate of fifty-three pieces per hour. The vertical spindle *C* is driven by bevel gears (enclosed in the guard *D*) and the pulley *E*, the latter being belted to a special countershaft. It is possible to drive this attachment without employing a special countershaft by connecting it directly through a telescopic knuckle-joint shaft to the gears driving the threading spindle, as illustrated in Fig. 5.

Another vertical-spindle slabbing attachment somewhat similar in construction to that just described is shown in Fig. 11. In this case, however, two end milling cutters A and B are held as shown. The spindles carrying the end-mills are driven from a special countershaft belted to pulley C. This pulley is keyed to the shaft D which drives the vertical shaft E through bevel gears enclosed in guard F. On opposite ends of

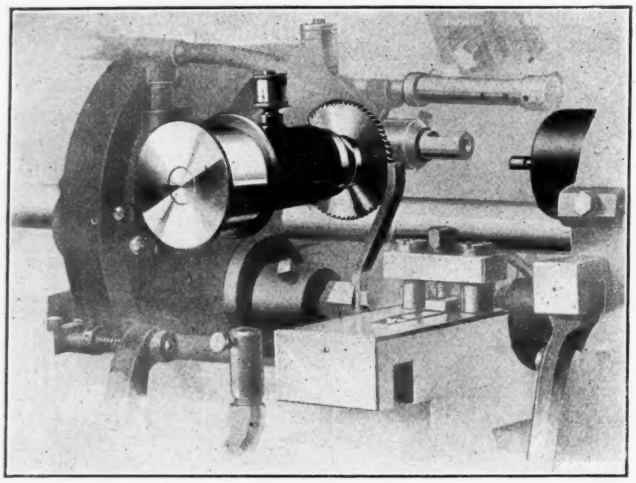


Fig. 15. Slotting or Milling Attachment held in Third Position Tool-spindle

shaft E are held gears G and H, which mesh with gears on the vertical milling spindles. This attachment is fastened to the top face of the cut-off tool-slide and is operated as previously described. Fig. 12 shows a modification of this attachment which increases its usefulness by the addition of the accelerating movement. This feature has been previously described.

A slab milling attachment carrying a face milling cutter having its axis parallel with the axis of the work is shown

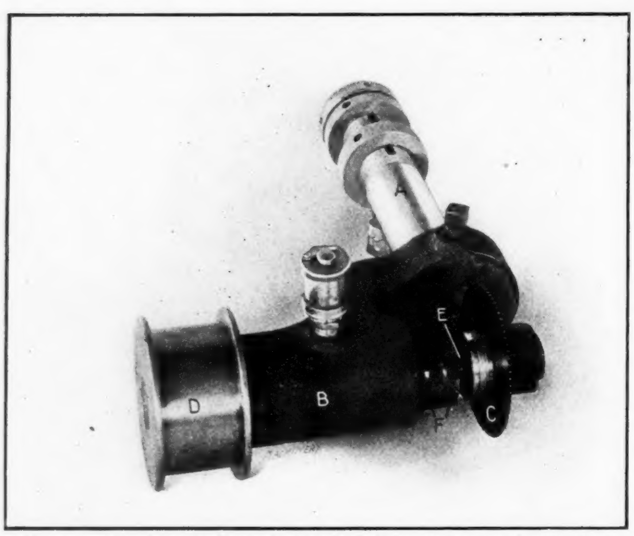


Fig. 16. Attachment shown in Fig. 15 dismantled from the Machine

in Fig. 13. This attachment is also held on the cut-off tool-slide, and is driven by pulley A, which is keyed to a shaft located at right angles to the milling spindle and carrying a bevel pinion that meshes with the large bevel gear B. This gear is located close to the cutter to obviate chatter and give a more powerful drive.

Another slab milling attachment which is suitable for heavier milling than that shown in Fig. 13 is illustrated in Fig. 14. In this attachment, the pulley A drives the milling cutter B through a worm C, worm-wheel D and two spur gears, not shown. The milling cutter in this case is 3½ inches in diameter by ⅞ inch face, and is serrated to break up the chips and facilitate the milling operation. A worm and worm-wheel drive is more powerful than a bevel gear drive.

Milling Attachment held in Main Tool-slide

An attachment which can be used for both slotting and end milling is shown in Figs. 15 and 16. This attachment is held in the threading spindle and is provided with a shank A, Fig. 16, to which the bracket B is held by a set-screw. The drive for the saw C (or milling cutter) is from a special overhead countershaft by a belt which runs on pulley D, the latter being fastened to the cutter-spindle E carrying the saw. The knurled nuts F are used to take up end play of the spindle.

Fig. 17 illustrates another type of end milling or slotting attachment, which is held in the third position and driven by gears. The bevel gear A receives power from the regular gears that are provided for driving the tools held in the second position tool-spindle. The cutter is adjusted for depth by means of a special device on the rear end of the main tool-slide. This attachment is held rigidly, being tied to both second and third position tool-spindles, and cannot be swung around to any position, as is the case with the attachment shown in Figs. 15 and 16. The work-spindle in the second

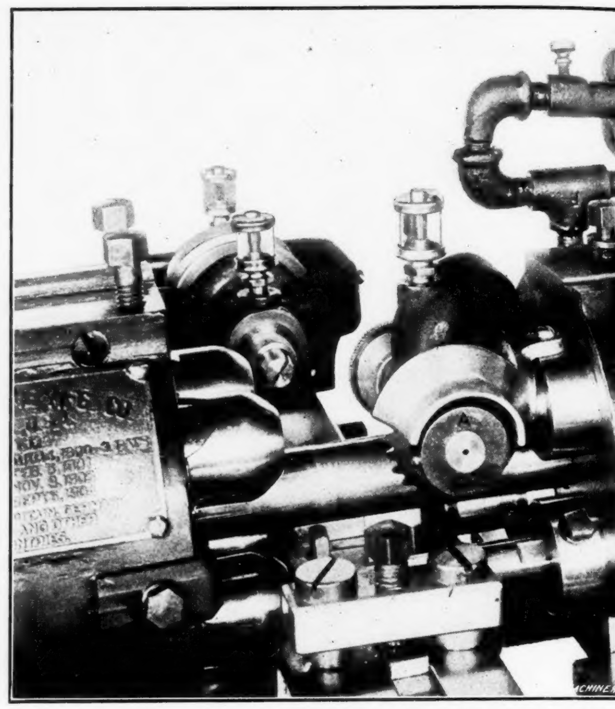


Fig. 17. Slotting or Milling Attachment held in Third Position and driven from Gears in Second Position Tool-spindle

position is stopped when the end milling or slotting operation is being accomplished.

The end milling attachment shown in Fig. 18 is also held in the third position tool-spindle, and driven by gears on the threading spindle through bevel gears in the attachment. In this attachment two cutters A and B are held on the vertical spindle, and are operated in a plane parallel to the top faces

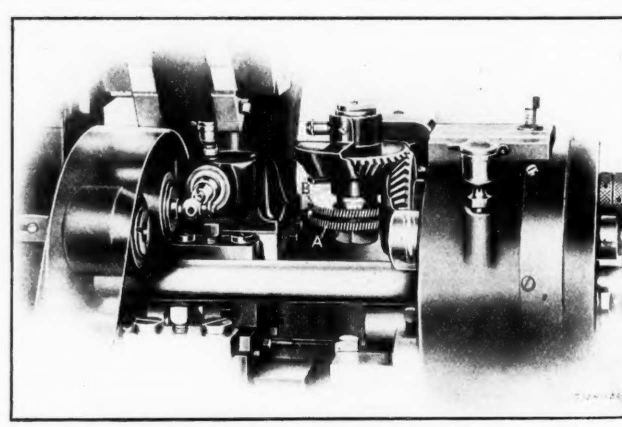


Fig. 18. Milling Attachment held and driven in Third Position Tool-spindle

of the side-working tool-slides. This attachment can be swiveled to any position for presenting the cutters in their proper relation to the work.

* * *

The British destroyer *Lurcher* achieved during its official trial of eight hours an average speed of 35.34 knots. The destroyer is driven by twin Parson turbines. The vessel is 265 feet long, 25 feet 7 inches beam. A speed of 35.34 knots corresponds to about forty-one statute miles per hour.

STANDARD TAPERS FOR SAFETY EMERY WHEELS

BY CHARLES G. SMITH*

The standardization of shapes for emery wheels and safety collars is a question which demands the immediate attention of manufacturers and users of grinding wheels. Unless a standard is adopted and complied with, the use of these collars and taper wheels will become more dangerous than safe and result in a number of accidents which might have been avoided. There are many companies using grinding wheels that have recently undertaken to make their own safety collars, and in several cases the collars have not been made to a shape which is in conformity with any of the established standards. This condition has been fostered by the fact that some manufacturers of grinding wheels are willing to make wheels to suit any shape of collar. This practice should be stopped immediately.

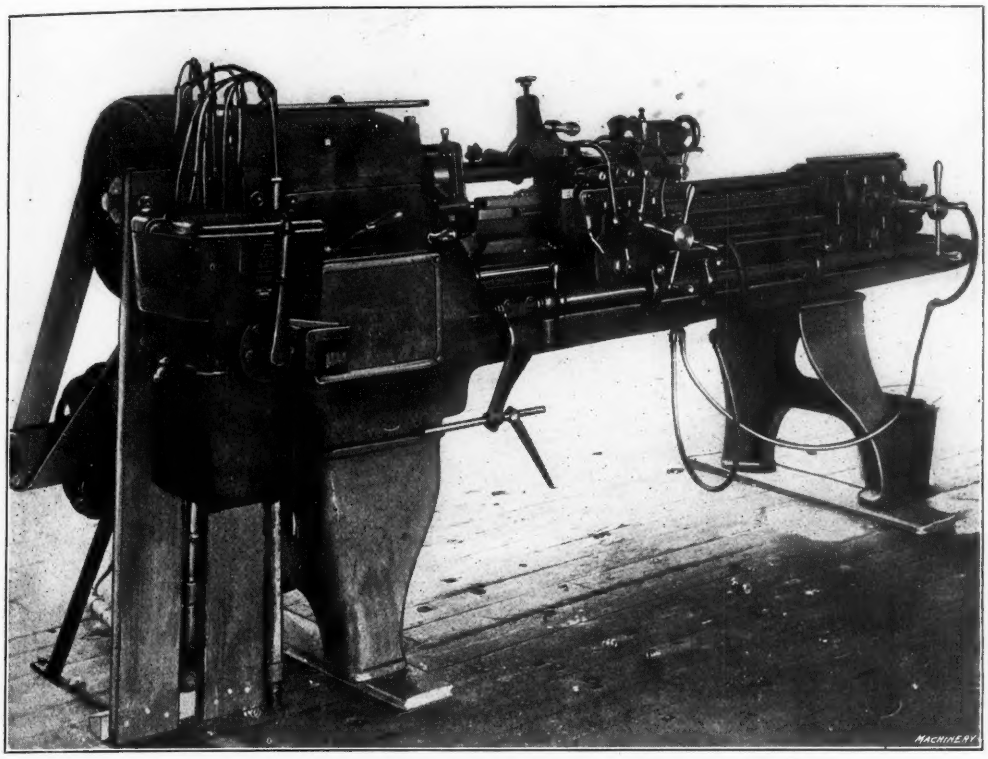


Fig. 1. A "Lo-swing" Lathe Motor-driven ready to turn Motor Armature Shafts

The Pittsburg Emery Wheel Co., Pittsburg, Pa., has adopted a standard taper of ¾ inch to the foot for grinding wheels and the safety collars which fit them, the wheels being tapered on both sides. Some of the older style of safety collars made by this company had the taper beginning at a distance of three inches from the center, but all of the collars are now made with the taper starting at two inches from the center, and all of the wheels have the untapered section at the center six inches in diameter.

The Safety Emery Wheel Co., Springfield, Ohio, formerly manufactured some of its safety collars with a taper of ½ inch to the foot, the taper beginning at a distance of two inches from the center, but several years ago a taper of ¾ inch to the foot with the taper beginning at a distance of two inches from the center was adopted as a standard.

The writer understands that the Carborundum Co., Niagara Falls, N. Y., has adopted a standard taper of ¾ inch to the foot, the taper beginning at two inches from the center of the collar in some cases and three inches in other cases. The wheels are made flat on one side and tapered on the other.

A number of instances have come to the writer's attention where pieces of a broken wheel have slipped from collars with a taper of ½ inch to the foot and caused serious damage. With the use of collars on wheels which are tapered on both sides to ¾ inch to the foot, where the collars and wheels fit each other perfectly and where the collar diameter is within four inches of wheel diameter, there have been practically no casualties or serious damage caused by broken wheels. By clamping a wheel with ¾ inch taper sides in a pair of collars having taper of ½ inch to the foot, the pieces are practically sure to fly in case of the wheel breaking, as the collars will not be in close contact with the wheel. A wheel with ½ inch taper sides will not necessarily break if mounted between collars with a ¾ inch taper, but the constant use of ¾ inch taper collars to hold a ½ inch taper wheel will ruin the collars for use on ¾ inch taper wheels, as the tightening of the nut in the spindle will spring the collars at the center. Where more than two inches of the wheel is exposed beyond the safety collars, a safety hood should be mounted on the machine, in addition to the use of the collars.

If all manufacturers of grinding wheels would adopt a taper of ¾ inch to the foot, with the untapered portion at the center of the wheel 6½ inches in diameter, and make all safety collars with a taper of ¾ inch to the foot, with the taper beginning at a distance of two inches from the center, it would appear that the maximum safety in grinding wheel operation would be assured. In this connection, it should, of course, be understood that users of grinding wheels can still make their own safety collars, but they should be made in conformity with the above standard. This standard applies to wheels of more than 12 inches diameter. Where wheels of less than 12 inches in diameter are used (this does not apply to 12-inch stubs of larger wheels) an untapered section three inches in diameter at the center of the wheel would be a satisfactory standard, and this would enable collars six inches in diameter to be used and still provide a sufficient hold on the tapered sides of the wheel when they were partially worn out.

* President, Pittsburg Emery Wheel Co., Pittsburg, Pa.

* * *

A "LO-SWING" LATHE TEST ON MOTOR ARMATURE SHAFTS

The Fitchburg Machine Works, Fitchburg, Mass., recently ran a series of tests to determine the motor horsepower required to turn electric motor armature shafts and the average time per shaft. The shafts are $24\frac{21}{32}$ inches long, 2 inches diameter, and are of 0.30 point carbon steel, being what is commonly known as "machinery" steel. The rough diameter of the stock was 2⅛ inches. The record of these tests is valuable, as they indicate the great economies that can be effected in machine shop practice by the use of "single purpose" machine tools developed for repetition work required to be machined in large quantities.

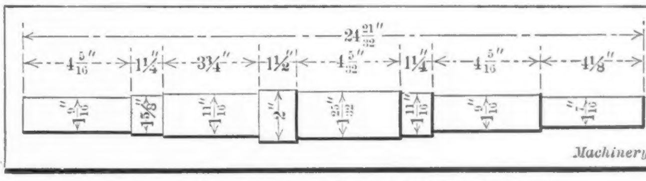


Fig. 2. Motor Armature Shaft of 0.30 Carbon Steel

The tests were run on a "Lo-swing" lathe equipped with the typical tool rests and turning tools used on this machine for shaft turning. The tool rests of this well-known machine tool are so constructed that there is only one slide between the tool and the lathe bed and, therefore, only one oil film between

the bed and the cutting tool. This rigid support for the tool enables the machine to take heavy cuts without chatter. The tool arrangement comprises multiple tool rests, all supported directly on the ways; these tools are adjusted at certain distances apart depending on the length of the different diameter sections of the shaft being turned.

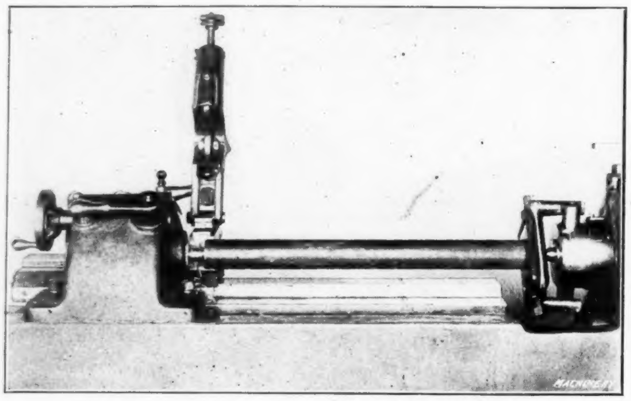


Fig. 3. Ready to start cutting with First Tool

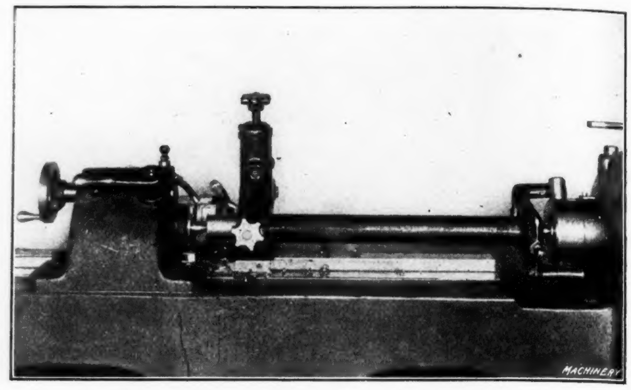


Fig. 4. Ready to start cutting with Second Tool

Fig. 1 shows the lathe as set up for the test. It is driven by a 10 H. P. Westinghouse motor, running at 1700 R. P. M., mounted on the back of the headstock leg and belted to the constant speed pulley. The controller is in front, convenient to the operator. The readings of power consumption were taken on an Esterline recording wattmeter which makes an autographic record consisting of a series of dots instead of the common unbroken line.

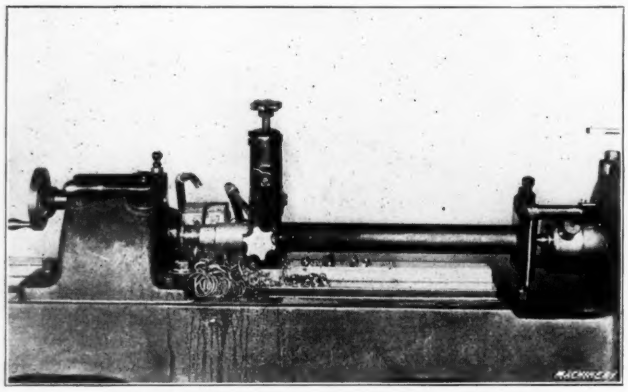


Fig. 5. Ready to start cutting with Third Tool

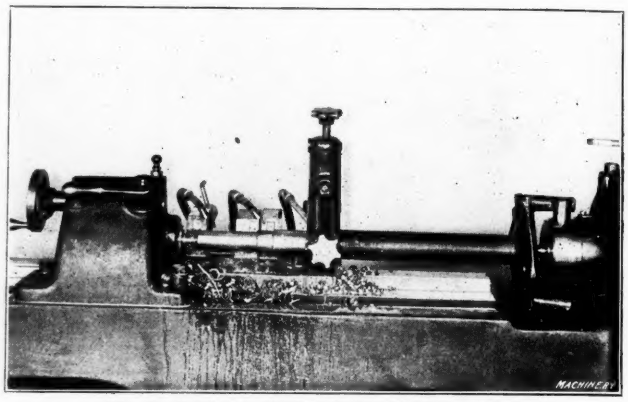


Fig. 6. Ready to start cutting with Fourth Tool

TABLE I. DATA ON SHAFT TURNING

Tools Cutting	Depth of Cut, Inches	Metal Removed per Minute, Cu. Ins.	Time Required per Cut, Seconds	Motor H. P.
No tools cutting......	0	0	0	1.2
Tool 1................	9/64	4.26	174	3 1
Tools 1 and 2.........	8/64	5.54	122	4.2
Tools 1, 2 and 3.......	1/16	7.14	106	5.2
Tools 1, 2, 3 and 4.....	1/16	8.62	52	6.3

Machinery

The turned shaft shown in Fig. 2 would require eight distinct turning operations if turned in an engine lathe with one tool. But on the "Lo-swing" lathe, the cuts on each end are taken progressively, four diameters being turned simultaneously when the tool cutting the smallest diameter goes into action. Fig. 9 shows diagrammatically the progression of the tools, the power consumption for each condition of turning being shown, that is, with one, two, three and four tools working, and the time required for each stage of cutting. The diagrams at the right in Fig. 9 show the horsepower for successive stages of turning and the time in minutes and seconds for each cut. The work speed was 250 R. P. M., or 131 lineal feet on the two-inch diameter. The feed was fifty turns to the inch.

The power required to overcome the friction of the motor and machine was 1.2 H. P. When the first tool went into operation, the power consumption rose to 3.1 H. P., the depth of cut being 9/64 inch, and the amount of metal removed 4.26 cubic inches per minute. When the second tool began cutting, the power consumption rose to 4.2 H. P., the metal removed then being 5.54 cubic inches for both tools. The progression of cuts and the corresponding power consumption and total metal removed per minute is given for one, two, three and four tools in Table I. Table II gives the corresponding data

TABLE II. DATA ON SHAFT TURNING

Tools Cutting	Depth of Cut, Inches	Metal Removed per Minute, Cu. Ins.	Time Required per Cut, Seconds	Motor H. P.
No tools cutting......	0	0	0	1.2
Tool 1................	1/32	1.00	127	2.0
Tools 1 and 2.........	5/32	5.54	107	3.8
Tools 1, 2 and 3.......	1/32	6.36	60	4.8
Tools 1, 2, 3 and 4.....	1/32	7.14	43	5.6

Machinery

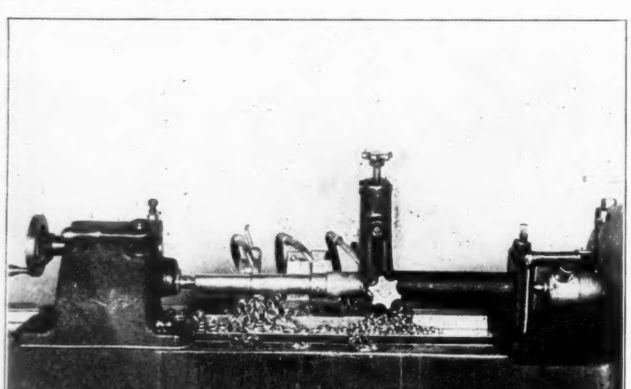


Fig. 7. Showing Four Cuts Finished

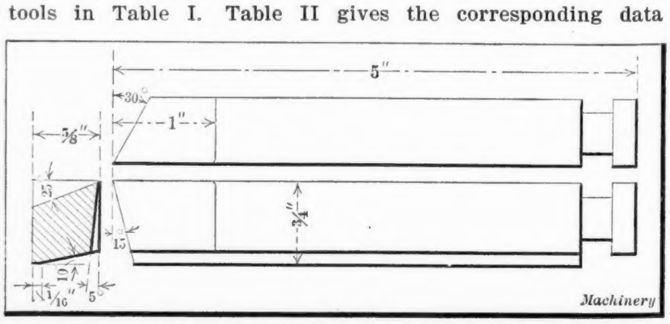


Fig. 8. Standard Turning Tool used in the "Lo-Swing" Lathe

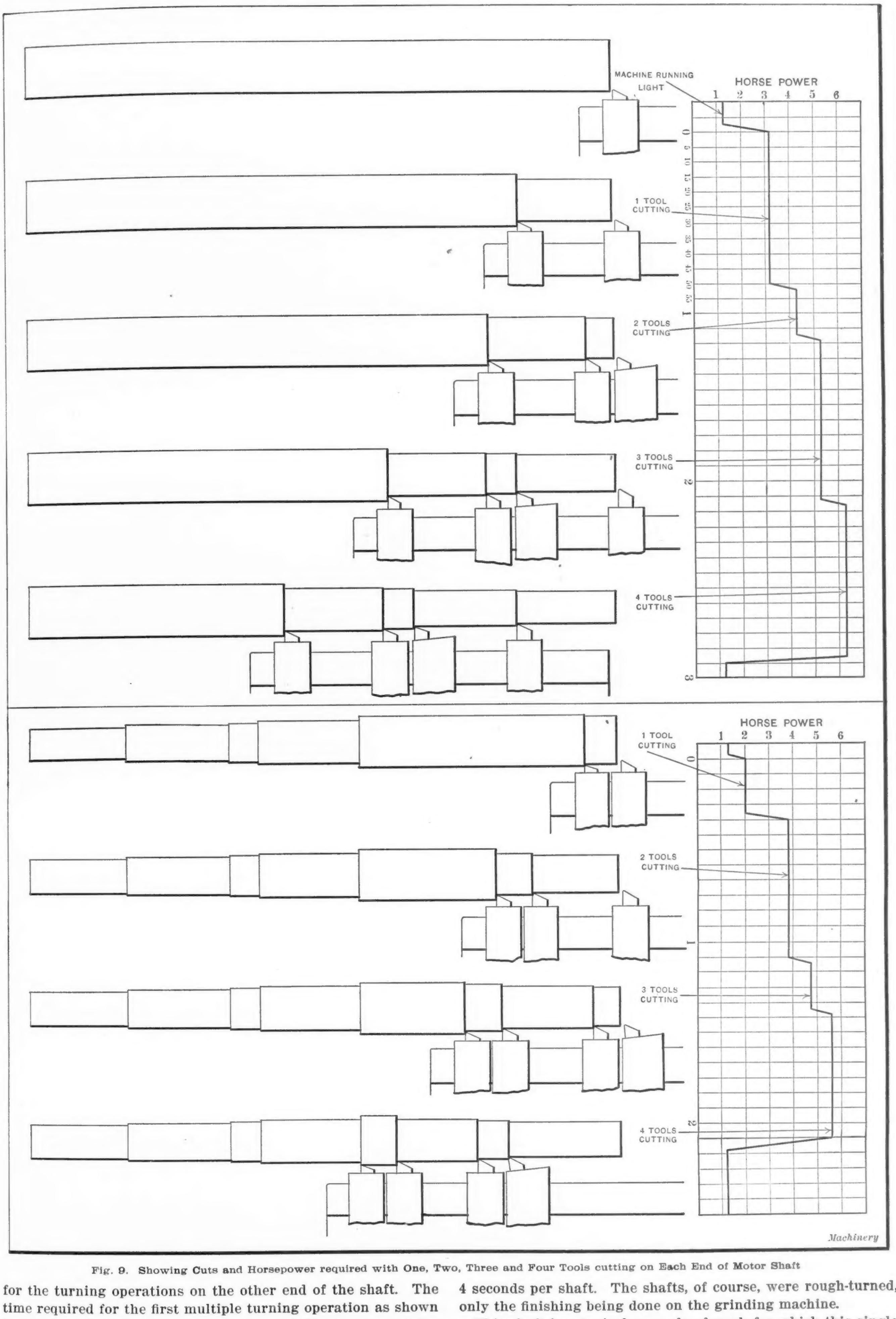


Fig. 9. Showing Cuts and Horsepower required with One, Two, Three and Four Tools cutting on Each End of Motor Shaft

for the turning operations on the other end of the shaft. The time required for the first multiple turning operation as shown in the diagram, Fig. 9, was 2 minutes and 54 seconds, and for the second multiple turning, 2 minutes and 6 seconds. The total time required for turning five shafts including the changes was 35 minutes and 20 seconds, or an average of 7 minutes and 4 seconds per shaft. The shafts, of course, were rough-turned, only the finishing being done on the grinding machine.

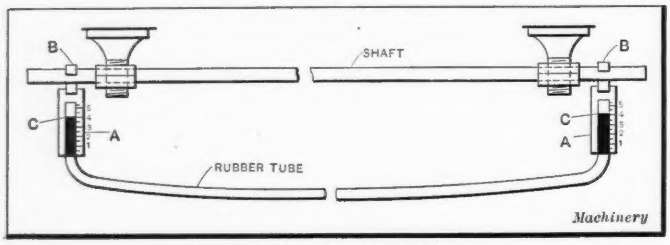
This shaft is a typical example of work for which this single purpose tool is adapted. The use of single purpose tools should become more general as their advantages over general purpose tools are more fully appreciated.

LETTERS ON PRACTICAL SUBJECTS

We pay only for articles published exclusively in MACHINERY.

METHOD OF LEVELING LINESHAFTING

In the February issue of MACHINERY Mr. E. L. White describes a method of leveling lineshafting. The writer has used a method somewhat similar to this but believes that it is better because it enables one man to handle the work without assistance. The accompanying illustration shows the apparatus required, from which it will be seen that it is along the same general lines as that described by Mr. White. The difference lies in the fact that the glass tubes at the ends of the rubber tubing are mounted on small wooden blocks A.



Method of leveling Lineshafting

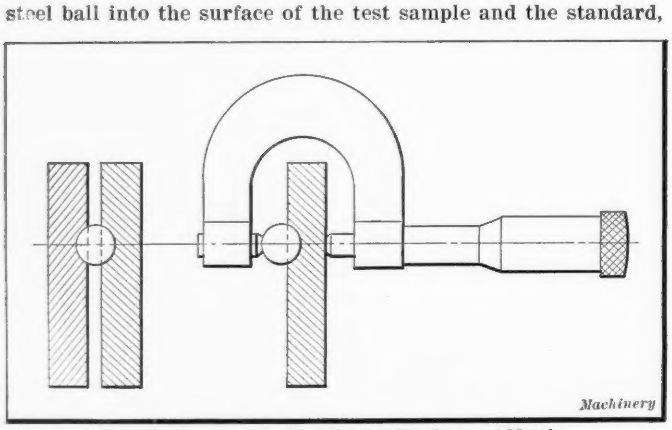
These blocks have a scale of inches laid off on them and are provided with band-iron hooks B which are used to hang the blocks on the shafting. The apparatus is filled with water, as described in the previous article under this heading, and in using it the two blocks are hung on the shafting at some distance apart. The level of the water at C is then observed in the two tubes and if there is any difference, it shows that the level of the shafting must be adjusted accordingly.

Youngstown, Ohio. H. ERSKINE

TESTING HARDNESS OF METALS

The accompanying illustration shows a method of testing the comparative hardness of the softer metals and alloys, such as bearing metals, different grades of brass, etc. The method is only comparative and it is necessary to have a standard with which to compare the sample that is to be tested.

Both the sample and standard plates should be flat and it is better to have them of the same thickness, if possible. The test is made by putting the two plates between the jaws of a vise and inserting a hardened steel ball between them. The jaws of the vise are then screwed up in order to force the steel ball into the surface of the test sample and the standard,



Test for determining Comparative Hardness of Metals

care being taken to have the pressure distributed uniformly on both of the plates. The next operation is to remove the plates from the vise, place the ball in the dent in one of them and caliper from the face of the ball to the back of the plate, as shown in the illustration. Then caliper the steel ball and subtract its diameter from the first reading. Subtract this result from the calipered thickness of the plate, and it will give the depth to which the ball has been forced into the plate.

If we let:

A = distance from the face of ball to back of plate,
D = diameter of the ball,
T = thickness of the plate,
B = depth to which ball was sunk into plate.

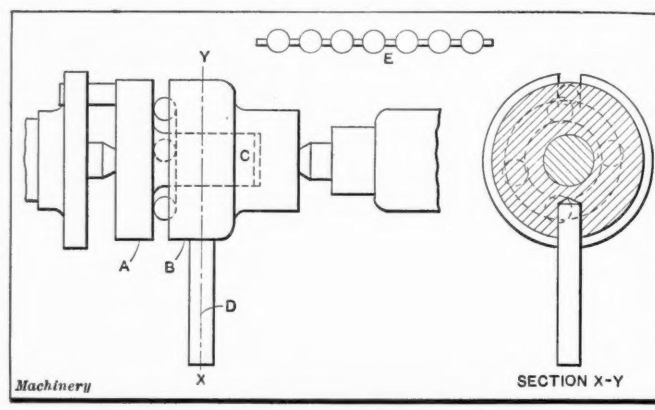
Then $B = T - (A - D)$

Perform the same operation with the second sample and then a comparison of the two values of B will give an approximate idea of the comparative hardness of the two metals under test. Of course, there are other factors that have some effect on the depths to which the ball will sink into the plates besides the comparative hardness of the two metals. But where only rough approximations are required, this method of determining the hardness of metals will give good results, and it has the advantage of being quick and simple.

Kenyon, R. I. H. M. NICHOLS

MAKING SOFT METAL BALLS

The device shown in the accompanying illustration was used by the writer for making soft metal balls. Referring to the illustration it will be seen that it consists of two members A and B which are mounted between centers in a lathe. The contact surface of plate A is flat and the balls to be formed are rolled between this surface and a groove in plate B. It will be seen that A is driven by a pin inserted in the faceplate, while B is prevented from rotating by the shank D which is held in the toolpost. The spindle C fits into a corresponding socket in B. The groove in which the balls run, is turned slightly eccentric to the center about which A rotates



Lathe Attachment for making Soft Metal Balls

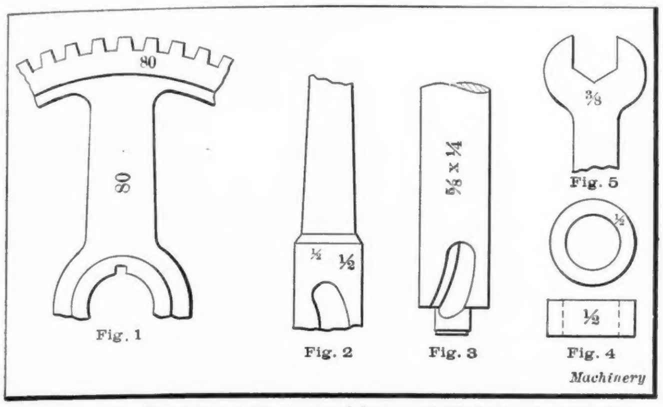
and, as a result of the crosswise motion imparted to the balls, combined with a steady and even pressure from the tailstock, an excellent finish is produced. The error in the finished work will not exceed 0.002 inch, which is quite close enough for the purpose for which these balls are required. The balls are cast in strings, as shown at E, and cut up to be finished in this attachment. Lard oil is used for lubricating the balls while they are being rolled. J. F. ESTLE

Linden, Mass.

NUMBERING MACHINERY AND TOOLS

The best method of placing numbers on tools and machine parts is not generally understood. The use of the proper sized numbers and their proper location is just as important a matter as any other machine shop operation. It is evident to one who has made any study of this matter that there are a great many machine parts on which the numbers have been misplaced. While it is important that all numbers placed on work should be as large as it is possible to have them, it is of equal importance for such numbers to be placed where they can be easily found. For instance, it is common practice to stamp the number of teeth on a gear on the side of the rim, as shown in Fig. 1. This is poor practice because the workman who is looking for a gear with a given number of teeth may have to look all around the rim before he locates the figures; this is especially true if the gear is "gummed up" so that the numbers are practically invisible. All gears should have the number of teeth cast on the arm or web, and where such a practice is followed the numbers can be found at a glance. Some tools are stamped with such small

figures that although they may be easily found they are nevertheless difficult to read. Fig. 2 illustrates a case of this kind where the advantage of using larger figures will be readily apparent. Fig. 3 shows the correct method of marking a counterbore, the diameter of both the cutter and the pilot being given. Fig. 4 shows two views of a collar used on a milling machine arbor; the top view illustrates the common fault of placing the figures on the side, where lack of space makes it necessary for them to be of small size. This method has a further serious fault, in that the figures are hidden when the collar is placed between two cutters on the arbor, so that there is sometimes an uncertainty as to its thickness. Both of



Figs. 1 to 5. Examples of Correct and Incorrect Methods of numbering Tools and Machine Parts

these faults can be removed by placing the figures on the edge of the collar, as shown in the lower view.

Although there is a great deal of work that is improperly marked, there is also a large amount which is not marked at all. When the sizes are not placed on tools, time is wasted in stopping to measure them. This results in the loss of time which is taken up in guessing and remeasuring to find the actual size of a tool. Wrenches form a simple example of this kind, but if all such tools had the size of the opening marked upon the face of them, as shown in Fig. 5, a lot of trouble would be avoided. Those who are interested in keeping down operating expenses in a shop will find that the time saved through the use of proper sized numbers and correct systems of numbering will constitute a considerable item.

Hartford, Conn. JAMES E. COOLEY

DISTRIBUTION OF CATALOGUES

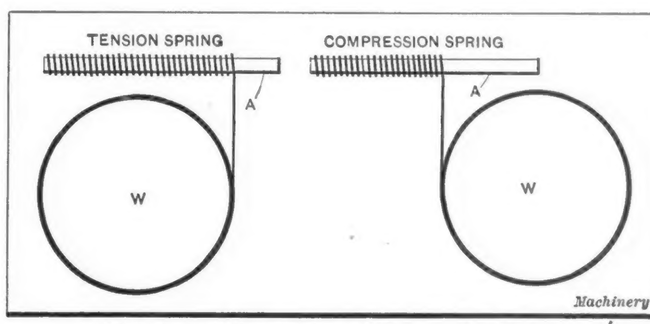
I was much interested in Mr. John F. Winchester's reply in the March issue of MACHINERY, to the editorial regarding the distribution of catalogues. During several years experience in drafting rooms and machine shops I have seen machinery catalogues put to a variety of uses, other than that of affording a reference for the purchasing department. The fact that nearly all machinery catalogues contain interesting data concerning the products which they show, and tabulated information on different technical subjects, would appear to indicate that manufacturers expect their catalogues to be used as a reference, knowing that this use will keep their products before the public. Mr. Winchester's plan of familiarizing himself with a given type of machine by studying the catalogues in which it is shown is certainly a good one. The average text books describe different classes of machinery, but do not often point out the features of different types as clearly as the manufacturer will do in his own catalogue. This fact will help to explain why so many students make requests for catalogues.

At the same time it cannot be denied that the liberality of different firms in regard to the distribution of catalogues is somewhat abused. I have seen a catalogue destroyed the same day that it was received. This was not through ignorance or neglect, but arose from the fact that, as there was no place provided in the factory for keeping catalogues, certain pages were torn out to be preserved in a data book and the remainder of the catalogue discarded. In this connection I would suggest that each firm supply its engineering department with a cabinet or bookcase in which to keep catalogues received from different manufacturers, instead of having such catalogues filed in the purchasing agent's office, where they are not available for the workmen to use. If this suggestion was followed, the men would have the use of larger and better bound copies from which they would be able to secure a considerable amount of valuable information. Many firms are now furnishing smoking and reading rooms for their men and no doubt a cabinet containing machinery builder's catalogues would constitute a valuable addition to the furniture of such rooms.

FLANGE

WINDING SPRINGS

The writer has seen a number of different methods of winding springs, but he has never seen any information given about the method of holding the wire. The accompanying illustration shows methods of holding the wire for winding tension and compression springs, respectively. In both cases, the wire is rolled on the spool *W*. In winding a tension spring, the spool is placed so that the wire runs off to the mandrel *A* from the right-hand side of the spool. The wire has a certain amount of set in it due to having been wound on the spool, and this set tends to bring the coils of the spring close together so that the spring is quite stiff. In winding a compression spring, the spool is held so that the wire runs off to the mandrel from the left-hand side. In this case the set in the wire will spring the coils apart, when the spring is taken off the mandrel, so that a compression spring is produced. The coils are always wound close together on the mandrel, and the tension or compression of the spring can be regulated by changing the size of the spool on which the wire is wound. This result can also be obtained by means of the lead-screw



Method of winding Tension and Compression Springs

on a lathe, but the writer knows of no way of winding a tension spring in which the coils are held together in so satisfactory a manner as in the method here described.

Westfield, N. Y. EARL W. JOHNSON

REASONS FOR THE RUNNING OF BELTS

I was interested in reading three recent discussions in MACHINERY on the "Running of Belts." George P. Pearce's assertions in the January issue, and J. R. Rand's in the February issue are substantially correct, but T. J. C's. assertion in the October issue is diametrically opposite to the others, and in my opinion incorrect. These contradictory assertions remind me of the following incident: A party of tourists were viewing Niagara Falls. "Isn't it wonderful!" they exclaimed. Overhearing this remark, a native replied: "There ain't nothin' wonderful about it; it's 'nater' for water to fall." This man knew that water falls, but he did not explain why. If asked why water falls, he would probably have answered, "Cause it does." The reason why water falls is that water is heavier than air. Everyone realizes this when the fact is brought to his attention. The reason why belts perform as they do is equally fundamental and easily understood when the simple facts governing belts are brought to the student's attention.

It is the purpose of this article to explain the self-evident facts that govern the performance of belts, and to express these facts so simply and clearly that the student may understand why belts perform as they do. He will then be able to regulate conditions to make them run as he desires.

A flat belt running on flat faced pulleys mounted opposite each other on shafts not parallel with each other, if placed on the edge of the pulleys nearer the ends of the shafts that are further apart, as shown by the original position *A*, tends to advance toward the ends of the shafts that are nearer together. Why? Because the belt is guided from each pulley to the other at an angle leaning in that direction. Under these conditions the belt will continue to advance toward the ends of the shafts that are nearer together with almost the precision of a nut running on a screw, until its tendency to advance in this direction meets an opposing tendency caused by the edge of the belt projecting beyond the edge of the pulley, as shown by the running position *A*. When the advancing edge of the belt projects beyond the edge of the pulley, the belt will automatically establish its running position where these opposing tendencies balance each other.

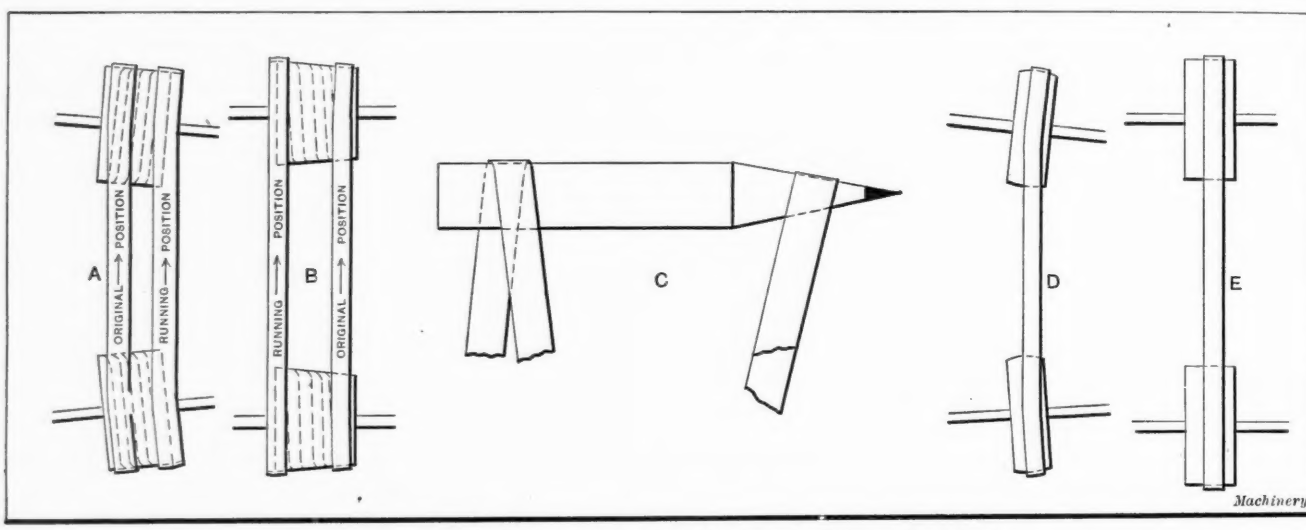
A flat belt placed on the smaller end of a cone-shaped pulley, as shown in the original position in diagram *B* tends to advance toward the larger end of the pulley. Why? Because the conforming of the belt surface to the surface of the cone-shaped pulley (see cone portion of diagram), causes the belt to be guided to the pulley face at an angle which leads toward the larger end of the pulley; also because the larger end of the pulley tends to cause the edge of the belt, there in closest contact, to travel faster longitudinally than its opposite edge, with the result that the belt is further skewed toward the larger end of the pulley. The belt will continue to advance toward the larger end of the pulley with almost the precision of a screw thread, as shown by dotted lines in diagram *B*, until its tendency to advance in this direction meets an opposing tendency caused by the edge of the belt projecting beyond the edge of the pulley. The projecting edge of the belt takes the same form and establishes the same conditionsas if the projecting portion were running on half of a cone-shaped pulley having the same taper as that established by the projecting edge of the belt shown in the running position in diagram *B*. Under these conditions the running position of the belt is established by the diameter of the pulleys; the distance between the shafts; the width, thickness, elasticity, and tightness of the belt; and the amount that the belt projects beyond the edge of the pulley.

A belt running on a centrally crowned pulley, equal in diameter at both ends, tends to establish its running position in the center of the pulley. Why? Because each half of the pulley is cone shaped, and the conforming of the belt of the pulley is cone shaped, and the conforming of the belt to the pulley face tends to guide each half of its width toward the center of the pulley. The belt establishes its running position on the pulley where these opposing tendencies balance each other.

A belt running on centrally crowned pulleys of equal diameter at both ends, and mounted opposite each other on shafts that are out of parallel, establishes a running position aside from the center of the pulley. The distance that the running position is off center depends upon the diameter of the pulleys, the amount they are crowned, the amount that the shafts are out of parallel, and the character and tightness of the belt. (See diagram *D*.) Why? Because the tendency of the crowned pulley to cause the belt to occupy the center of the pulley is partially overcome by the tendency of the belt to advance toward the ends of the shafts that are nearer together. The belt establishes its running position where these opposing tendencies equally balance.

A belt that is perfectly straight, flat, and uniform, running on pulleys equal in diameter for their entire width, having faces wider than the belt and mounted opposite each other on shafts that are parallel, will maintain the running position which it occupied immediately after finding its first balance, if it is not forcefully guided from its path. (See diagram *E*.) Why? Because the belt is guided from each pulley to the other at perfect right angles.

To establish and maintain these conditions—perfect belts, pulleys and alignment of the shafts—all of which are necessary to the perfect running of belts, is practically impossible. Therefore, as average conditions are only approximately perfect, pulleys must be slightly crowned to assure a satisfactory operation of the belts running over them. Pulleys should not, however, be crowned more than is necessary, because excessively crowned pulleys destroy belts and waste power. Why? Because excessively crowned pulleys force the belt to become V-shaped in cross-section. When such a belt leaves the pulley, the center and edges of the belt are constantly being stretched and compressed. This action ruins the belt and causes practically the whole width to slip on the pulleys, which action wastes power.



Diagrams illustrating Reasons for running of Belts under Different Conditions

Several of the assertions made in this article may be easily, clearly and conclusively illustrated by winding a strip of paper or tape around the cylindrical or sharpened cone shaped portion of an ordinary lead pencil. Even so simple an experiment is sufficient to prove that the guiding of belts is the law which governs their performance.

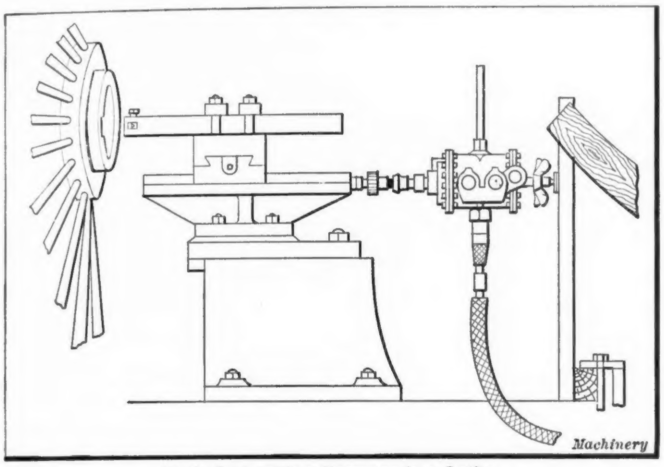
Hartford, Conn. ARTHUR I. JACOBS

CUTTING KEYWAYS IN A LATHE

The accompanying illustration shows a rig which the writer got up to cut keyways in three-rope pulleys, 16 feet in diameter. These pulleys were bored out on a faceplate lathe; the bore of the pulley was 12 inches in diameter, and the keyway was cut 3 inches wide by 11/16 inch deep. The length of the hub was 29 inches. After boring out the hole and facing off the hub, an air drill was connected to the screw of the tool-rest in the manner shown in the illustration. The pulley was still mounted in position on the faceplate of the lathe and a square nosed tool about 5/16 inch in width was placed in the tool-holder. The air drill used to drive the feed-screw was of the reversing type, and a boy manipulated the throttle so that the direction of the rotation was reversed when the tool reached either end of its stroke, thus giving a good imitation of the reciprocating stroke of a

slotter. It was necessary to have a boy for this work because the crank of the cross feed was too far away to allow the machinist to handle the reversing of the drill. The drill was operated under 100 pounds air pressure at the compressor, and the cut which it was possible to take in this manner was really surprising.

Before the development of this method we had always cut keyways of this kind by hand, as we did not have a keyseating machine and the slotter in our shop was too small to handle work of this kind. It required about eight days of our best mechanic's time to cut one of these keyways by hand, while the present method enables the work to be completed in less than two days. It is obvious that the keyway will be square with the bore of the pulley because the same setting of the work on the lathe faceplate suffices for both



Method of cutting Keyways in a Lathe

the boring and the keyseating operations. In order to cut a keyway by hand and still have it square requires particular skill on the part of the mechanic working with a hammer, chisel and file.

New Glasgow, N. S. JOHN S. WATTS

TUBE BENDING DIE

In the manufacture of incandescent lights, we had a number of delicate tube-bending operations to perform, such as shown in Fig. 1. The difficulty of these operations lay in the fact

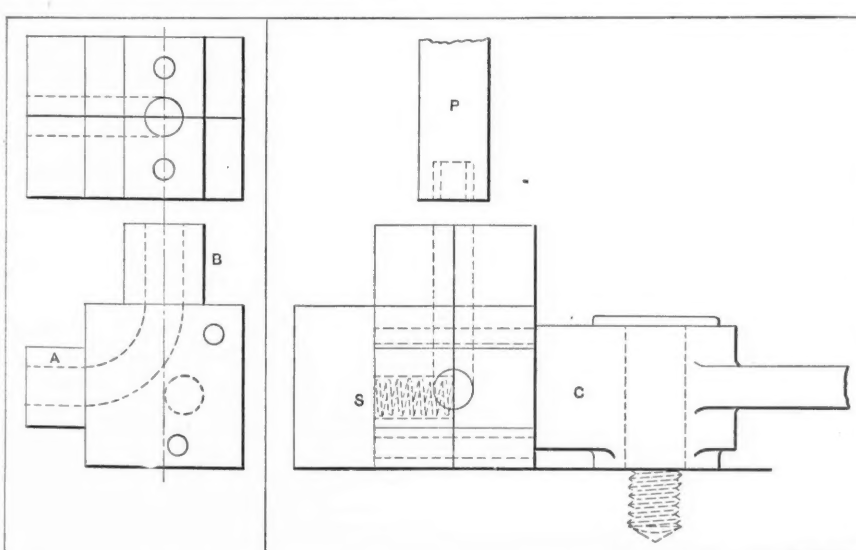


Fig. 3. Finishing Die ready to be mounted in the Press

Fig. 4. The Punch and Die showing Method of opening to remove Work

that the tubes had to be perfectly smooth and free from wrinkles. Several different methods were tried, but none of them proved quite successful until the die shown in Figs. 3 and 4 was developed.

The first step in making the die is shown in Fig. 2. A circular groove was turned in a plate, the width being made equal to the diameter of the tube and the radius equal to the radius to which the tube was to be bent. This plate was parted along line X–X and one half was again cut along line Y–Y. The two quarters of the plate produced by the second cut were fitted together with dowel-pins as shown in Fig. 3. As the two ends of the tubing had to be perfectly square, the guides A and B were added to the die. These guides consist of two blocks, through which holes equal in diameter to that of the tube were drilled; the blocks were then split and

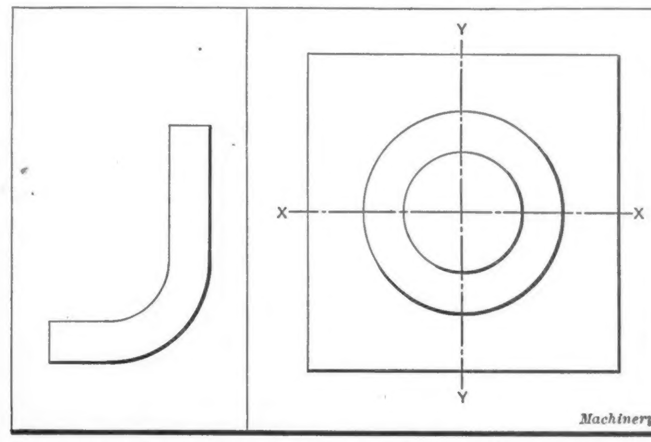


Fig. 1. The Form to which the Tubes had to be bent

Fig. 2. First Step in the Production of the Die

fastened in position as shown in the illustration. A hole was drilled in one block to receive the spring S.

Fig. 4 shows the assembled die. In bending a tube, the blank is put in at the top of the die and then forced down by the punch P. It will be seen that the provision of a cam C, turned by a hand lever, provides for releasing the two halves of the die so that they can be forced open by the tension of spring S. The finished tube can then be readily removed. This method of bending proved very successful; the tubing came out perfectly smooth, and without wrinkles, and the ends were absolutely square. F. X. BUCHER

Jersey City, N. J.

KNURLED NUTS

Knurled nuts on machines and tools do not always seem to be applied in the right place. It is understood that the use of pliers on knurled surfaces is bad practice, yet they are being used in the shop every day to secure a tighter grip on a nut than can be obtained by hand. The knurl might be protected by a piece of brass while using this method, but so many men do not do this that the nut is soon spoiled. Therefore provision for a wrench should certainly be made when the knurled nut cannot be made large enough to enable it to be properly tightened by hand.

Detroit, Mich. FELIX HEBERLEIN

TESTS MADE WITH "STELLITE" TOOLS

I noticed a brief article by Mr. William Schellenbach, in the March issue of MACHINERY, entitled "Comparative Chip Production of Stellite and Steel Tools." Mr. Schellenbach made these statements with perfect fairness and described quite clearly the results achieved by steel tools in a large lathe. I will say regarding tests made with "Stellite" tools that the object was not to determine the amount of metal removed in a given time, regardless of other conditions, but to determine the amount of metal that could be removed in a given time under stated working conditions. In every instance the tools were comparatively small — usually $\frac{3}{8}$ inch square. In all cases where rapid cutting is of supreme importance, the depth of the roughing cut is predetermined by the amount of stock to be removed. That is, the roughing cut must remove practically all of the surplus metal, leaving the piece ready for the finish-

ing cut or for the grinder. Under these conditions there remain only two methods of increasing the amount of metal to be cut from the piece in a given time. First: by increasing the feed; second: by increasing the speed. It is by the best combination of these two factors that a given tool is enabled to remove its maximum amount of metal (limited by the quality of the tool) in a given time. If the lathe is not so constructed that it will carry the work under this feed and speed, the tool is naturally unable to perform its full duty.

A practical illustration will probably make this matter clearer. An operator was engaged in turning pistons. With a steel tool he was able to complete the roughing cut in two minutes and six seconds; while with a "Stellite" tool, he finished the same cut in 54 seconds. Now there was no way in which any steel tool could take this cut in much less than two minutes and six seconds, because the edge of the steel tool would be destroyed by an increase of either speed or feed. The "Stellite" tool was only able to remove this cut in 54 seconds because two men applied pressure to the four-inch belt in order to keep the lathe in motion until the cut was completed. Both the speed and the feed were increased. There are cases, of course, in which it becomes a question of removing the greatest amount of metal in a given time, *regardless of all other conditions*, and I can only say that, so far as I know, "Stellite" has never been tested on this particular kind of work. The bar from which the cuts mentioned by Mr. Schellenbach were taken was of a large diameter, and it seems certain that the depth of the cuts must have been considerable; this would necessitate a large tool working under heavy pressure. It would be interesting to know the exact conditions under which this remarkable test was made. Mr. Schellenbach has certainly shown that the lathe achieving these results was a very powerful machine.

Kokomo, Ind. ELWOOD HAYNES

FLANGING BRASS TUBING

The tools shown in the illustrations were designed for use in flanging the ends of pieces of brass tubing to fit them onto similar tubing at an angle of 80 degrees. In order to make these joints fit properly, the ends of the tubing have to be cut to such a shape that a flange could be produced which would fit evenly around the joint. The first cut on the end of the tubing is made with the tool *A*. The end of the tubing

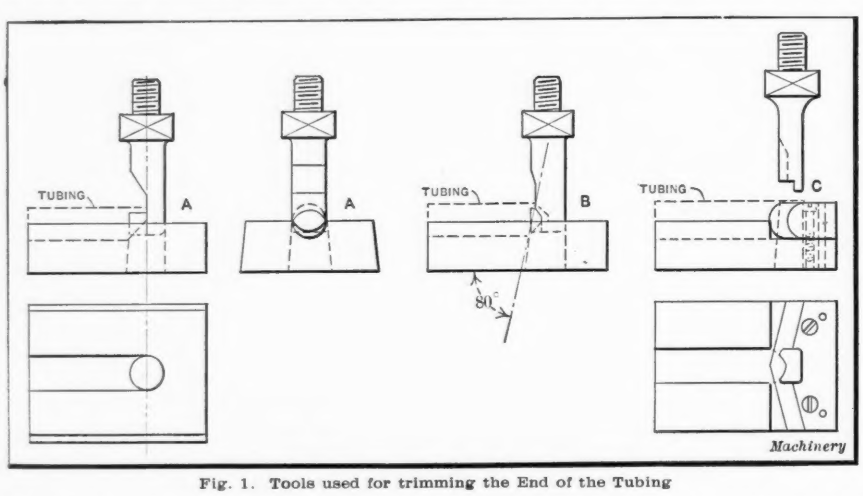


Fig. 1. Tools used for trimming the End of the Tubing

is slipped over the elliptical shaped extension on the punch, and when the press is tripped, the lower half of the end of the tubing is sheared to the required shape. The wall of the tube is comparatively thin so that it does not require much pressure for the cutting operation. It will be seen that the upper side of the tubing bears against the punch so that the latter forms an end gage. The tubing is then turned over and a second cut is taken with the tool *B;* this tool is similar to the first one except that the punch is not cut away so much, in order to allow the end of the tube where the first cut was taken to abut against the punch and bring the tubing into the proper position for the second cut. In other respects the punch and the operation performed by it are similar to the punch *A* used for the first operation. It will be evident that the end of the tubing now has two sharp corners where the ends of the two cuts taken in the preceding operations meet. These corners are trimmed off by means of the tool shown at *C;* this tool is arranged so that the second corner may be trimmed off by simply turning the tubing over. It will be seen that the tool is provided with an end stop for locating the position of the cut, and as both cuts are gaged

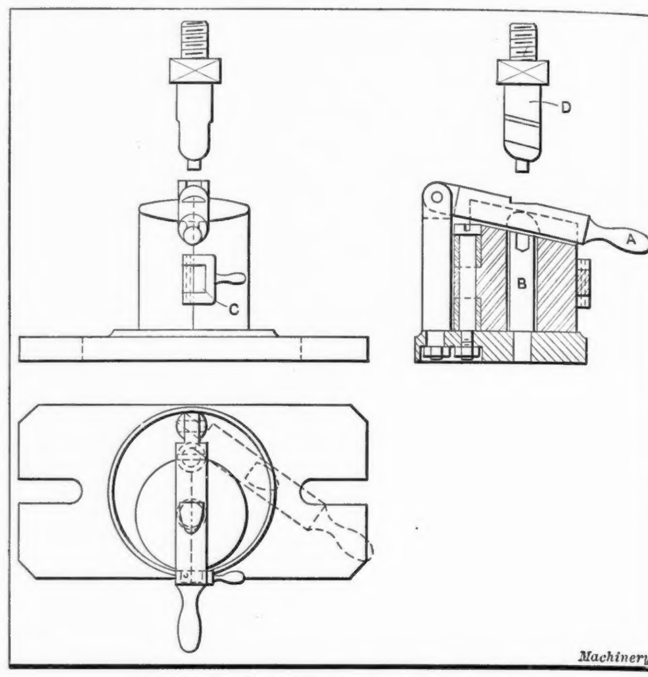


Fig. 2. Punch and Die for forming the Flange

from the upper side, it will be evident that the corners are trimmed off to equal lengths.

After these trimming operations have been completed, the work is ready for the flanging operation. The tool shown in Fig. 2 is used for this purpose. It consists of two hinged jaws, bored out to fit the tubing which they close around. The center piece *B*, which is turned to a sliding fit in the tubing, is mounted so that it is in a central position between the jaws when the latter are closed. When closed, the jaws are held together by means of the latch *C*. The lever *A* is hinged to an upright piece and can be swung to one side, when necessary, as shown by the dotted lines. This lever locates the tube in the die and also constitutes the finishing punch for the flanging operation. The flange is started with the punch *D*, the operation being as follows: A piece of tubing is placed over the center *B* and the lever *A* is brought down onto its end to bring it into the desired position; the jaws are then brought together and fastened by means of the latch *C*. The lever *A* is now swung to one side and the punch *D* is brought down to start the flange; the stroke of the punch *D* is regulated by a stop set on the press. The lever *A* is then placed in position on the tubing and the punch is again brought down on top of this lever. The jaws are hollowed out so that the flange is finished to the required shape between the space in the jaws and the cylindrical surface of the lever *A*. The work produced in this way was entirely satisfactory, the flanges being perfectly smooth and of an equal width.

Meriden, Conn. JAMES GALLIMORE

ERASING INK MARKS ON TRACINGS

The writer has noticed a number of letters in MACHINERY describing different methods of erasing ink lines from tracings, and in this connection he wishes to say that he has found a sheet of blotting paper particularly useful for this purpose. When it is discovered that a mistake has been made before the ink has time to dry, the greater part of it

can be removed with the blotting paper. This leaves only a faint line which can be erased with an ink eraser, leaving the surface of the cloth in good condition. If the ink has had time to dry, it is, of course, a somewhat more difficult matter to remove it. This may be done very successfully with a sharp knife, taking care not to cut the cloth. After the greater part of the ink has been removed in this way, the erasure can be completed with an ordinary rubber on a lead pencil. The writer has used this method for a number of years and can recommend it as being worthy of trial.

Cincinnati, Ohio. E. C. McMeans

CORRECTING DRAWINGS

An excessive amount of work is sometimes necessary when making changes in a design, or rectifying errors on a part that has already been traced. In order to reduce the time necessary to make such changes to a minimum, the following method will be found to give excellent results:

Fill the ruling pen with water and go over the various lines to be effaced—in the case of dimensions use an ordinary writing pen instead of the ruling pen—then proceed to erase. You will be surprised, after you have tried this method, to find that the time consumed in making changes will be very materially reduced.

J. P.

METHOD OF MACHINING ELLIPTICAL CASTINGS ON A LATHE

A plan and sectional view of an elliptical casting for a man-hole cover is shown in Fig. 1. It was required to machine these castings on the flange *A*. The shape of the work made it necessary to devise some special means of machining.

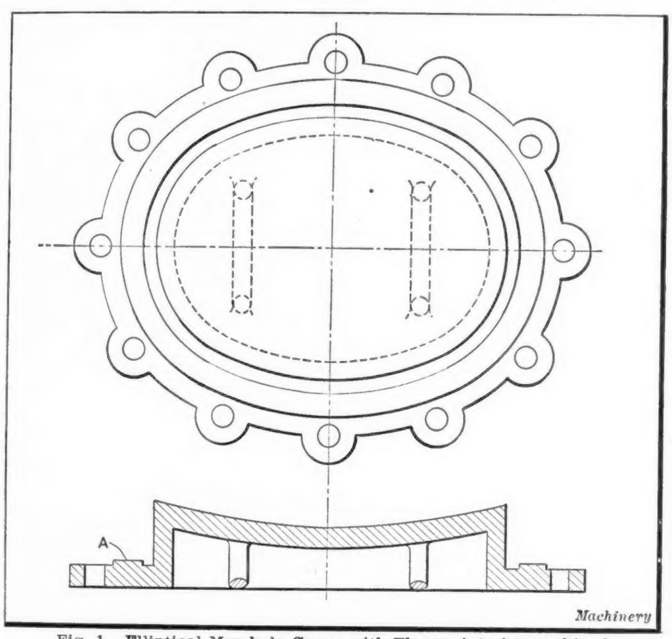


Fig. 1. Elliptical Man-hole Cover with Flange A to be machined

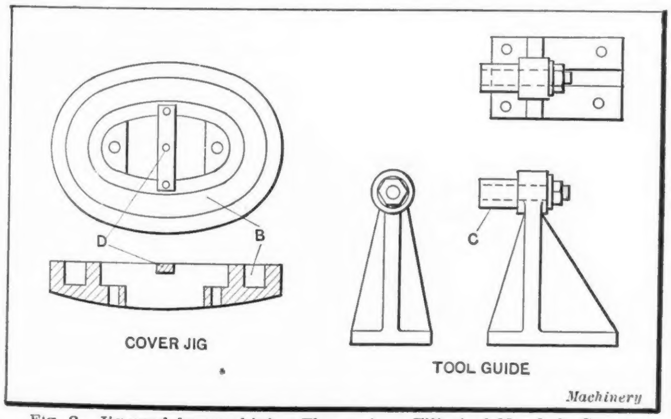


Fig. 2. Jig used for machining Flange A on Elliptical Man-hole Covers

Fig. 2 illustrates the jig which was finally used. It consists of two main parts—the cover jig with an elliptical slot *B* in it, and the tool guide carrying roller *C* which engages in slot *B*. The cover jig is bolted onto the man-hole cover, the latter being bolted to the faceplate of the lathe after centering it by means of the hole in the bar *D* on the center of the cover. The roller frame is bolted to the tool carriage, as shown in Fig. 3, and as the work revolves the roller travels in the elliptical slot *B* and draws the tool back and forth so that it is kept on the flange *A* of the man-hole cover all the time. The same method could be applied in a variety of other classes of elliptical work which have to be turned on a lathe.

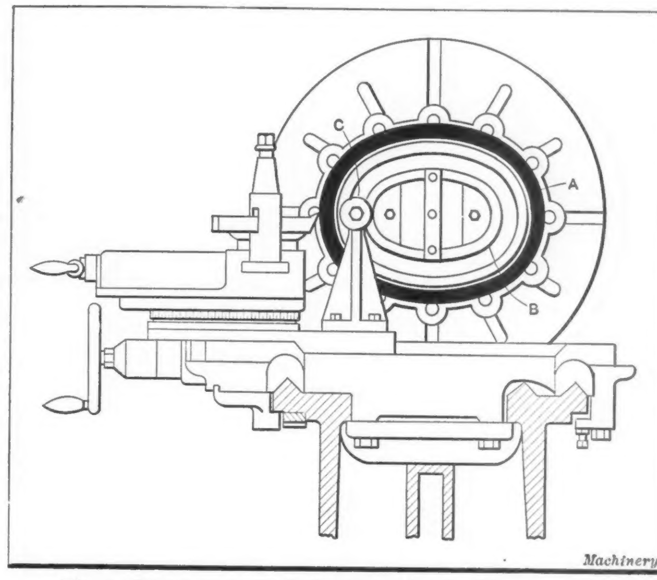
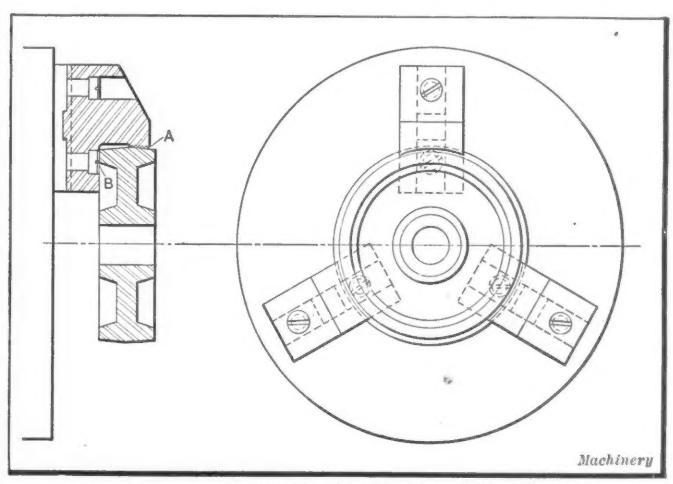


Fig. 3. Jig set up on the Lathe for turning the Flange A on the Man-hole Cover

Philadelphia, Pa. C. W. Carrigan

CHUCK JAWS FOR HOLDING DROP-FORGINGS

In the February issue of Machinery, an article was published describing a special form of chuck jaws for holding drop-forged gear blanks. In attempting to judge the value of these jaws, one is working somewhat in the dark as the grip depends upon the manner in which the piece is to be machined and this was not explained in the article. The accompanying illustration shows a special form of chuck jaw for holding work of this kind. These jaws are made of case-hardened machine steel and grip the work on the taper *A*. This tends to pull the work toward the chuck where it is backed up against the rim *B*. The chuck jaws are scored to prevent the work from slipping when the cut is started.



Chuck Jaws designed for holding Drop-forged Gear Blanks

Two operations are involved in machining these gear blanks. The first operation is as follows: The work is gripped on the outside diameter and the hole is drilled, bored and reamed. The side of the rim and hub are then faced. The work is then ready for the second operation, for which the work blank is placed on an expanding arbor; the face of the gear is then turned, after which the side of the rim and hub are faced. This completes the lathe work.

Rhode Island

FORMING TOOLS FOR ROUGHING AND FINISHING

In the January issue of MACHINERY Mr. Corwin Lamoreaux described a circular forming cutter for manufacturing ball-bearing cones, and as the writer has had some experience with these wide tools he wishes to criticise a few of the claims which were made for a combination roughing and finishing tool. Considering cutters of this style from the toolmaker's standpoint, they can always be made cheaper and better in two parts owing to the difficulty of cutting the clearance in the groove *A* Fig. 1. This is especially true when this groove is narrow and required to be more accurate than the one described in Mr. Lamoreaux's article. When made in two parts, the sections of the tool are fastened together with screws. From the operator's viewpoint these wide cutters are trouble-makers, as the work heats and binds in the groove, causing from 1/4 inch to 1/2 inch of the tool to be wasted owing to the temper being drawn. For this reason two cutters of simple design, as shown in Fig. 2, will give better results. The cut-off tool *B* forms the hub of the work and holds it accurately to length. The use of the wide tool requires an allowance by the cut-off tool for facing, and this will cause trouble when the cut-off varies longitudinally, as shown in Fig. 1. This condition will cause the work to be formed with a shoulder, and the same variation in cut-off is responsible for the hub of the next piece being too short, although the operator may not suspect that there is any trouble, on account of the form tool apparently facing at *C*.

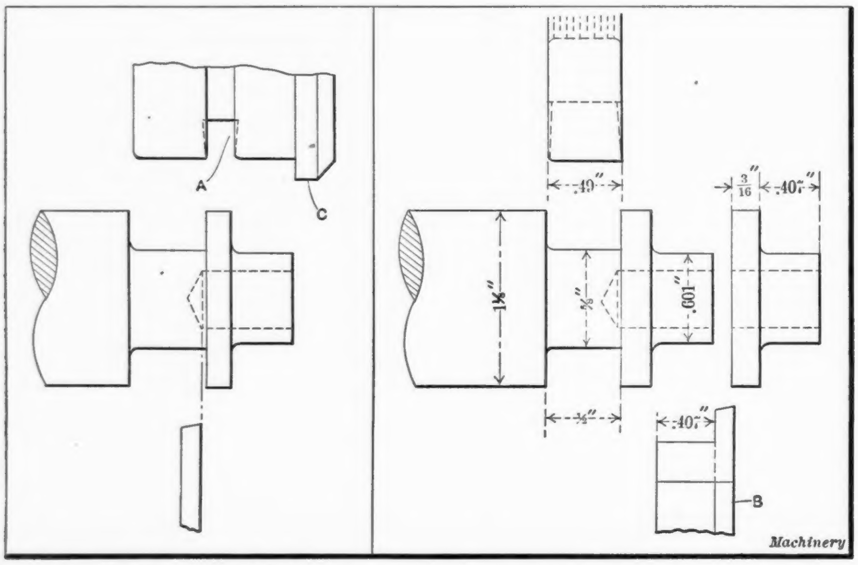


Fig. 1. Wide Form of Tool with Roughing and Forming Sections in One Piece

Fig. 2. Two Separate Tools for performing the Same Operation

The layout in Fig. 2 allows both tools to cut simultaneously at one-half the feed per revolution allowed for the wide tool, and in either case the work should be cut off where the stock is 1⅛ inch in diameter and not where it has been reduced to ⅝ inch in diameter, as the cut-off tool is likely to scar the work while being brought to the forward position. In regard to the rate of production, the separate tools will produce over 100 per cent more than the wide tool, when used in automatic machines, as they may be operated simultaneously. This relation of production is shown in the following table.

Number of Revolutions Required with Wide Tool

Form: 0.262 inch travel at 0.0005 inch feed	524 revolutions.
Cut-off: 0.375 inch travel at 0.001 inch feed	375 revolutions.
Total	899 revolutions.

Number of Revolutions Required with Two Tools

Rough form: 0.255 inch travel at 0.001 inch feed (while cutting off)	255 revolutions.
Cut-off: 0.375 inch travel at 0.001 inch feed	375 revolutions.
Total	375 revolutions.

The method of facing described by Mr. Lamoreaux should give better results than when facing from the turret, as in the latter case the cutter must be kept ground to 90 degrees and it is apt to show tool marks. It is not usually considered good practice to drill while forming with a very wide tool, owing to the excessive heat of the cut and the pressure of the wide tool causing the drill to cut large; furthermore the speeds are not always suitable for forming and drilling simultaneously.

ELAM WHITNEY

WEAR IN SHAPER RAM GUIDES

The writer has reason for supposing that the usual method of gibbing the shaper ram is by no means so bad as A. H. makes it appear in his contribution in the November, 1912, issue of MACHINERY. According to his dictum, "where the wear is great, the design presents a great drawback on account of the difficulty of adjustment of the upper strips." Again, objection is raised to the initial cost of fitting and adjusting the slide on a new machine. Moreover, A. H. asserts that the lack of adjustability is a bad feature when the slide has become worn through use.

All these arguments, with the exception of that relating to initial fitting, which is largely a matter of opinion, would be just if the wear of both slide units were uniform throughout their length. But this is not so, as will be readily apparent after a careful examination into the direction of the forces at the cutting edge of the tool, of the forces at the propelling mechanism, and of the gravitational forces. The last named forces result in pressure on the lower surfaces of the ram and on the upper surfaces of the shaper body. At the front end of the slide, the downward pressure is intensified by those cutting forces which tend to pull the tool into the work, and such a condition usually results in "digging in" and consequent vibration. These forces may wholly counteract the gravitational forces at the back end of the ram, thus causing the separation of the bearing surfaces and preventing wear on the bottom surfaces, but intensifying the wear on the upper surfaces. Again, under certain conditions, the tool receives a side thrust which tends to tilt the ram. In fact, conditions vary to such a degree that it is practically impossible to locate the points subject to greater wear with any degree of accuracy. It is, however, absolutely certain that the wear cannot be uniform, and there are points in the slides where the wear is practically negligible. The surfaces of both slides tend to wear rounded. The ram slides are subject to little wear at the ends; in fact it might almost be said that no wear develops at the extremities of the ram slides, since they seldom make contact with the opposing body slides. Personal experience has shown that attempts made to adjust almost any form of slide will result in failure. If the gibs are raised to take up the wear in the frequently used portions of the slide, the absence of wear in the portions that are used less frequently causes binding.

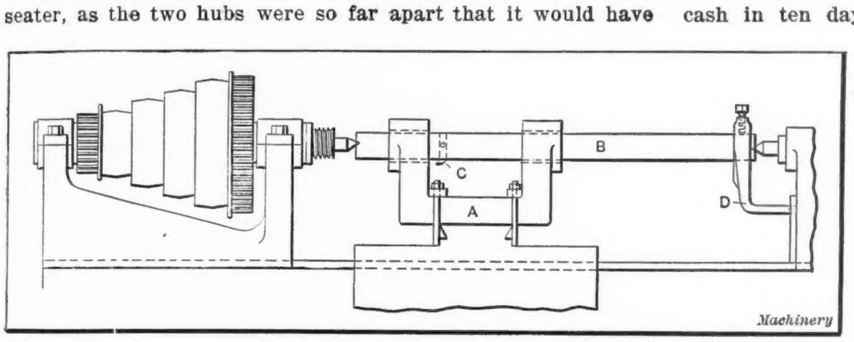
What, then, is the inference when the subject is considered from this point of view? Professor Sweet has stated that surfaces which tend to uniform wear do not wear at all, or, at any rate, are subject to little wear. His remedy to secure this desideratum was to examine the parts after running under normal conditions, then to bodily remove those subject to no wear, and to reduce the area of those subject to little wear inversely as the amount of the wear. This is not at all a bad remedy, providing the conditions of service are known. In general purpose machines, however, conditions are so variable that this method is impracticable. The presence of the shallow grooves produced in removing metal to reduce the surfaces would also permit the penetration of grit between the sliding surfaces. The evident conclusions are: (*a*) It is impracticable to secure uniform wear. (*b*) Certain parts are subject to practically no wear. (*c*) Consequently, the provision of adjustments for wear is unnecessary. This does not debar incorporation of adjustments for the facilitation of construction.

The writer has often been struck by the meagreness of the sliding surfaces in the shaping machine, as usually constructed. Is there any good reason why these surfaces should not be increased? With the surface areas at present in vogue, readjustment is necessary under ordinary conditions of service at least once a year if the quality of the output is to be maintained. Now suppose that all the surfaces were at least doubled in area. Would it not be certain that their durability would be at least twice as great, rendering necessary but a biennial adjustment? In fact, when one considers that present surfaces are so limited as to be within the "cutting" region, it might be stated that the life would be lengthened even more than 2 to 1 by the increase. Wider surfaces would tend to retain the oil, preventing metal-to-metal contact. What the writer wishes to press home is this: If portions of both sliding surfaces do not wear, adjustment can only be effected by scraping; hence, means provided for adjustment other than those convenient for the initial assembly are useless. Even the taper or parallel gib might be dispensed with, if this afforded no convenience in erection. FRANCIS W. SHAW

Manchester, England

CUTTING A KEYWAY IN A LATHE

A simple but novel way of cutting a keyway in a lathe is shown in the accompanying illustration. The work is shaped like the letter U and a keyway is to be cut in each hub. This job could not be conveniently accomplished in the keyseater, as the two hubs were so far apart that it would have been necessary to reverse the work to cut the second keyway.



A Makeshift Device for cutting a Keyway in a Lathe

In the illustration the work *A* is bolted to the saddle of the lathe, while a boring-bar *B*, carrying a cutter *C* and being prevented from turning by the dog *D*, is held on the centers. To cut the keyway, the back-gears are thrown out and the cone is loosened on the spindle, so as to feed the carriage without revolving the spindle. Of course the feed-nut was engaged with the lead-screw and the carriage traversed at the required feed. This is only a makeshift, but it answers the purpose where other tools are not available. W. ALTON

SECTIONAL DRAWINGS

In the February number of MACHINERY, J. M. Henry writes in regard to cross-hatching sectional views of an assembled drawing. To be sure, it is a tiresome method, but is not all labor more or less tiresome? In regard to the use of tinting ink applied to the reverse side of a tracing, the writer believes that unless the tracer is an artist the result will look more like an oil painting by "A. Dauber" than the tracing of a drawing. Also the bluish-white color for representing sections on a blueprint is likely to cause trouble, for that same bluish-white may be produced by a tracing which has become spotted with water—a thing which happens quite frequently where the sun exposure method of printing is used.

The writer is employed in a shop where a great deal of special machinery is built, so that he has had ample opportunity to examine blueprints made in other shops; and if the tinting method should become standard practice, a great number of these prints would show nothing but sectional views. It has been the writer's experience, that the pencil layout and details are clearer to the detailer and tracer when sections are represented by different colored crayons and the final tracing is neatly cross-hatched, for the expense of the tracer is practically nothing, compared with the results, if that print is to be used to help sell a machine.

Dexter, Me. E. W. TATE

IMPROVED GAGE FOR TWENTY-NINE DEGREE THREAD TOOLS

In the February issue of MACHINERY, Everett Chipman describes a gage which is theoretically an improvement, but the writer trusts that the author does not believe that it is commercially possible to produce a hardened worm that does not vary more than 0.001 inch. He also states that it is a common error to set the top of the thread tool parallel with the axis of the worm. The writer would like to ask if the same assertion would prove true in the case of U. S. standard and square threads, and if Mr. Chipman could draw the outline of a tool that would be used in cutting a worm or square thread of 6- or 8-inch lead?

Dexter, Me. E. W. TATE

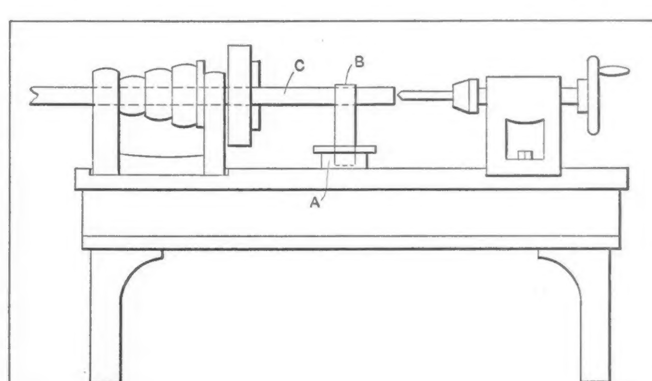
COMPARISON OF SELLING METHODS

In the February issue of MACHINERY, Mr. J. F. Winchester comments on my article on the subject of "Selling Methods," which was published in December. He asks whether thirty, sixty or ninety days is not considered cash payment and whether we use "twentieth century methods" in offering to pay cash. There is a discount of two per cent offered for cash in ten days, which is always taken advantage of by "live" concerns, and it was this kind of cash payment that I made reference to. Some people may consider it a cash payment when they settle their bills in one, two or three months, but most firms make a practice of allowing an extra cash discount to those who discount their bills within ten days of their presentation. In the case mentioned, our firm was willing to pay for the machine at once, but the sellers wished us to use it for ninety days and then pay for it at the usual discount, and if anything in the machine proved unsatisfactory, it could be shipped back to them at their expense. This is the reason for differentiating between a cash payment and a ninety day trial offer.

New Britain, Conn. J. M. HENRY

COOLING A DEEP-HOLE DRILL

The following is a method of cooling a deep-hole drill which is not original with the writer although he has never seen it used before. Any mechanic will readily understand this method by referring to the illustration, which shows a small



Method of cooling Work when drilling Deep Holes in a Lathe

pan *A* that is set on the lathe shears and partly filled with water or oil. An endless belt *B* dips into this pan—a woven belt will be found most satisfactory for this purpose, as it has a greater power of absorbing the liquid—and as the belt rotates water or oil is carried up onto the work and keeps it cool during the drilling operation. The pan can be moved along the lathe bed so that the belt will run above the point

where the drill is working. The writer saw this method in use in drilling deep holes in copper bars; this work formerly became exceedingly hot, but the application of this method enabled the heat of the cut to be so effectually dissipated that no trouble was experienced. WHIP

A LATHE CENTER OILING DEVICE

The illustrations show a useful device for oiling a lathe center, which consists of a cylindrical piece *A* screwed onto the center and having another cylindrical piece *B* revolving in it. This piece is made to revolve by means of a hard rubber washer *C* which is fastened to it by four countersunk screws. When the oiling device is screwed onto the center, this rubber washer is pressed against the work, and is carried around by friction. A ring *D* is turned on the inside of the sleeve *B*; this ring keeps passing through the oil or lubricant and carries a portion of it around and drops it on the center. The revolution of *B* is facilitated by means of four small rollers *E* which fit in a groove turned in the sleeve. These rollers are carried by studs as shown, and in addition to the reduction of friction which they effect, they serve the purpose of making the oiling device self-contained when it is not in position on the lathe center. The rubber washer also tends to keep the oil from leaking out when it is pressed in position against the work. *F* is an oil hole for refilling the device when necessary. A modification of this design could probably be made for use in different classes of work.

Manchester, England. W. R. OAKES

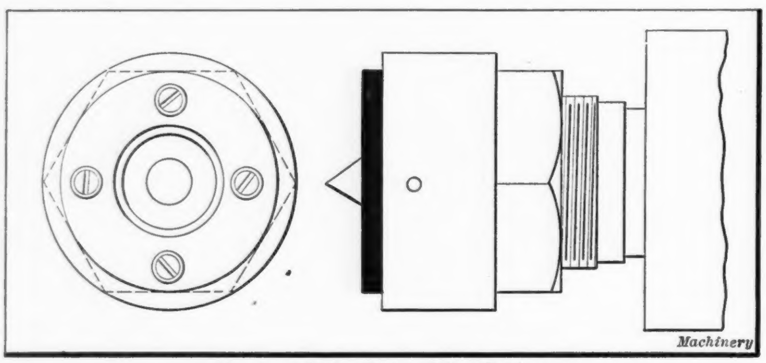


Fig. 1. A Lathe Center Oiling Device

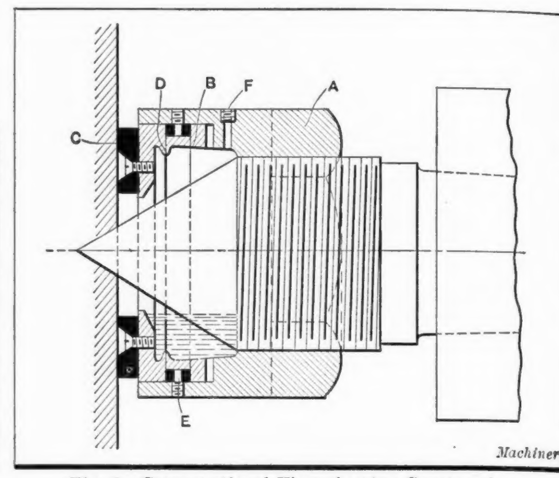
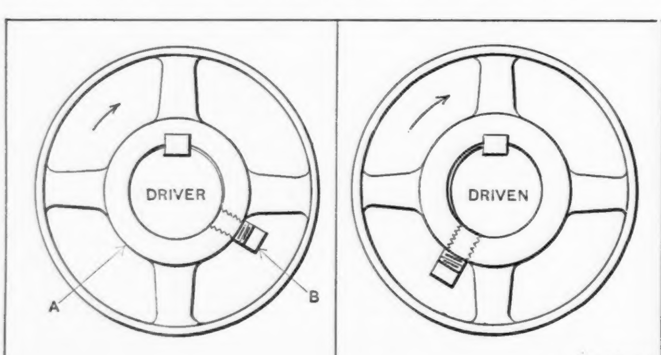


Fig. 2. Cross-sectional View showing Construction

LOCATION OF KEYS AND SET-SCREWS

Pulleys and gears are customarily secured to the shaft by means of a key and set-screw as shown in the accompanying illustration. That it makes a great difference in stability to have the set-screw on the proper side of the key is sometimes overlooked, and the accompanying diagram is shown to make clear how they should be located. Any hub that is not forced on its shaft with some pressure and that is secured by a driven taper key, or by a set-screw, or by a straight key with set-screw on top of it and a second set-screw at ninety or one-hundred twenty degrees from the key, assumes the position shown exaggerated in Fig. 1; *i. e.*, the hub is forced away from the shaft at the key and at the set-screw and bears hard on the shaft at the point *A*. Now if this shaft is a driver through the pulley in the direction indicated, it is in the position it would assume when power is applied, if no set-screws were used. As the key is the only connection between the pulley and the shaft in such a case, the pulley would swing to the left to whatever extent the bore would allow. It being desirable, however, not to have any lost motion between the pulley and shaft, the set-screw shown at *B* serves the useful purpose of preventing lost motion, and it is not called upon to do more than this. The hub assumes a wrapping position around the shaft, being secured to it by the key, and has a bearing for the full length of the hub at the key and also at the point *A*. Now if this shaft is rotated in the opposite direction, the same wrapping action takes place on the opposite side of the key, with the result that nothing but the point of the set-screw and the key have a bearing on the shaft. When put together in this way, the set-screws will soon wear into the shaft and become loose, and there will be lost motion between the pulley and the shaft which will inevitably cause trouble. For such a case, the set-screw should be at the point marked *A*, while the pulley will bind at the point where the key is shown. Fig. 2 shows the proper position for the key and set-screws where the pulley drives the shaft in the direction indicated.

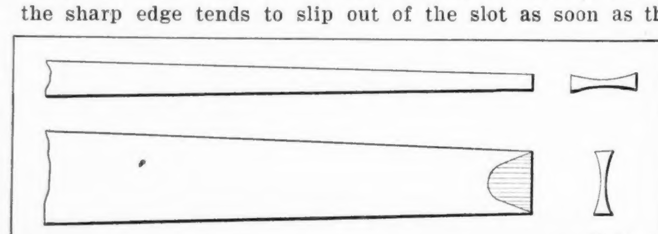
Watervliet, N. Y. MARTIN H. BALL



Proper Location of Keys and Set-screws for Different Directions of Rotation

FITTING A SCREW-DRIVER FOR UNIVERSAL SERVICE

It is a common sight to see the point of a screw-driver ground down to a knife edge, in an attempt to make it take the place of different sized tools for use in various sized screw slots. The practice of grinding a screw-driver in this way, generally means that it is ruined for efficient work, because the sharp edge tends to slip out of the slot as soon as the screw begins to turn hard. This spoils the slot in the screw head.



Screw-driver fitted for Universal Service

A good way to fit the point of a screw-driver to make it suitable for use on various sized screws is shown in the accompanying engraving. This illustration shows four views of the point, from which it will be seen that the center has been slightly hollowed out so that it will take small sized screws, while enough metal is left at each end of the point to give it sufficient size and strength for driving home large sized screws. The method will be readily understood, as the small sized screws have narrower heads and narrower slots, and require a thinner edge to enter them. This method of grinding does not impair the efficiency of the screw-driver nor the strength of the point for use on larger sized screws.

Upton Works, Mich. L. J. WATSON

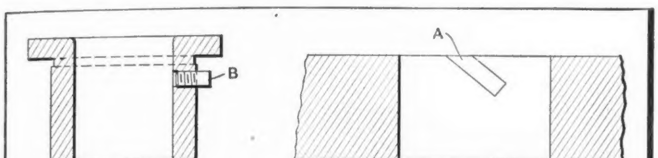
SHOP KINKS

PRACTICAL IDEAS FOR THE SHOP AND DRAFTING-ROOM

Contributions of kinks, devices and methods of doing work are solicited for this department. Write on one side of the paper only and draw sketches on separate sheets.

TO HOLD SLIP JIG BUSHINGS IN PLACE

The writer has found the following method for holding slip jig bushings in place a very good one: Drill and tap a ¼- or ⅜-inch hole in the side of the jig bushing, as indicated in the engraving. After the bushing is hardened and ground, screw in a pin and cut it off, so that it projects about 3/16 inch outside of the bushing, as at *B*. Chip out a slot in each hole in the jig as indicated at *A*, the hole being chipped in the direction of a spiral. By engaging the projecting pin in this



slot, the bushing is prevented from turning and from rising out of the hole. At the same time it can be easily gotten out when required, and there is no projection on the jig of any kind that can be broken off while handling. The writer has used this method on very heavy jigs.

It is not always necessary to tap a hole for the pin in the jig bushing. A plain drilled hole is sufficient when the bushing is at least ⅜ inch thick. If the wall of the bushing is thinner than this, the pin cannot be driven in tight enough to stay in place securely.

Akron, Ohio H. C. Armstrong

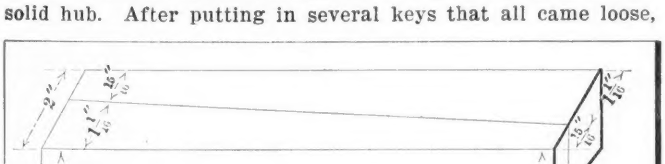
A HARDENING KINK

When hardening stamping dies or other tools having holes or sharp corners and projections, it is sometimes necessary to fill the holes and sharp corners with fire-clay to prevent the cracking of the tool in the bath. When the tool is put into the furnace, the clay dries and shrinks away from the steel, and sometimes drops off before the piece is removed from the furnace. A satisfactory method for overcoming this difficulty is to apply the clay as usual, and then heat the article enough to dry the clay somewhat, so that it starts to shrink; then some thin fireclay is applied to the tool as a paste to cement the clay to the steel. This fireclay should be of about the consistency of cream. A little practice will give good results.

Aurora, Ill. C. Lamoreaux

KEY FOR A REPAIR JOB

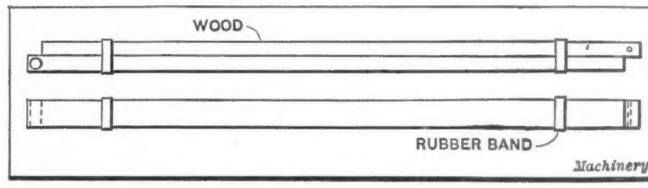
The writer has had to take care of the repairs of a 50-horsepower gas engine, provided with a clutch and driving a lineshaft connected to a flange on the flywheel, which had a solid hub. After putting in several keys that all came loose,



a key was made as shown in the accompanying engraving, and this key has met the requirements. One part of this key was first driven in fairly tight, and then the other part was driven alongside of it, thus providing a tight fit the entire length of the keyway. W. K. M.

HOME-MADE BEAM COMPASS

The following is a description of a home-made beam compass which worked very satisfactorily. Two pieces of soft pine, ⅜ by ¾ by 16 inches, were fastened together by two strong rubber bands, thus allowing any desired adjustment. A hole was drilled in the end of one piece to receive the leg of the dividers, while another hole was drilled in the other piece to receive either pen or pencil as desired. The results were

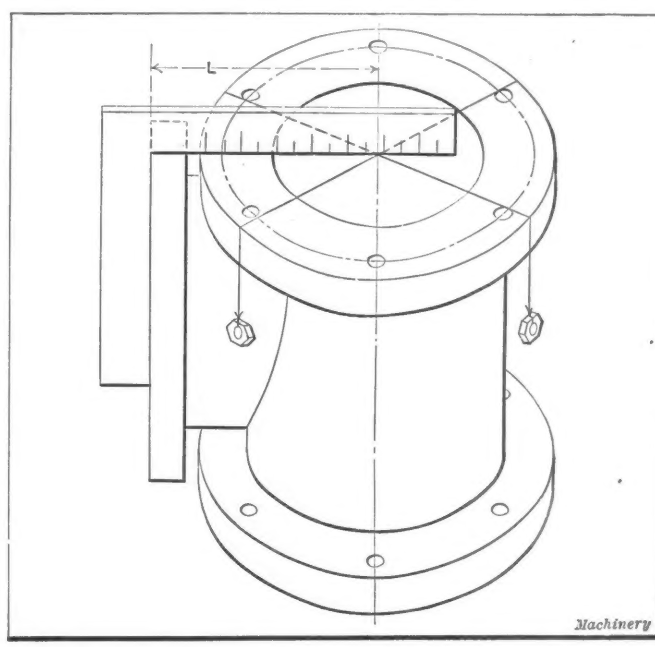


fully as good as with a high-priced instrument, and the advantage over the ordinary home-made beam compass was in the large range and the accurate adjustment that it was possible to obtain.

Erie, Pa. Thorvald Petersen

MEASURING RIGHT-ANGLED FITTINGS

Probably no person accustomed to sketching right-angled fittings has failed to find difficulty in getting the distance from the face of one branch to the center of the other, particularly if there are no facing strips. Usually it is impossible to measure to the bolt circle, as the holes are generally drilled off center. Both the inside of the fitting and the rim of the



flange are left unfinished, and unless one goes to the trouble of inserting a wooden center, the measurement is largely a matter of guess-work.

The following method has the advantage of great accuracy, and permits of reading the measurement directly. Place one of the faces in a horizontal position. Lay two threads, weighted at the ends, tangent to opposite holes, as shown in the engraving, being careful to keep to the same side of the holes all the way around. The intersection of these threads will then be the center of the face. By holding a square against the other face, the required dimension can be read off directly. This method, of course, cannot be used when the number of holes is uneven. V. C. Blackett

Glace Bay, Nova Scotia

TURNING A PIECE TO TWO DIFFERENT DIAMETERS

A piece turned from the solid to two different diameters, making a square shoulder, will be most liable to break in the corner. By making the piece in two parts, breakage may be prevented. A sleeve the same size as the large diameter is driven or shrunk onto a piece of the same diameter as the smaller part to form the shoulder. This eliminates the sharp corner or starting point of a fracture.

Norwich, Conn. George W. Armstrong

HOW AND WHY

DEPARTMENT INTENDED TO CONTAIN CORRECT ANSWERS TO PRACTICAL QUESTIONS OF GENERAL INTEREST

Give details in full and name and address. The name and address will not be published with the answer.

IMPRESSIONS FOR BOSSES IN DROP-FORGING DIES

O. L. B.—Why are the impressions for bosses on drop-forgings preferably put in the upper die?

A.—Because an impression in the upper die fills with metal more readily than in the lower die. The reason for this has been ascribed to inertia of the metal being formed, but this seems to be a hardly adequate explanation. Explanations from readers are invited.

POWER REQUIRED TO DRIVE PROPELLERS

H. R.—Is there a formula for finding the horsepower required to drive a propeller of a certain pitch and diameter? For example, suppose the diameter is 28 inches, the pitch 29 inches, and that the propeller is driven at 500 R. P. M.

A.—A formula is given by Kent in his "Mechanical Engineers' Pocket-Book," as follows:

$$D = 20{,}000\sqrt{\frac{\text{I. H. P.}}{(P \times R)^3}}$$

in which

D = diameter of propeller in feet;
I. H. P. = indicated horsepower;
P = pitch of propeller in feet;
R = revolutions per minute.

This formula can be transcribed into the form:

$$\text{I. H. P.} = \frac{D^2 \times P^3 \times R^3}{20{,}000^2}$$

From this we find by inserting the values given in the example above:

$$\text{I. H. P.} = \frac{2.33^2 \times 2.42^3 \times 500^3}{20{,}000^2} = 24\text{, approximately.}$$

DEFLECTION OF ANGLE IRONS

L. M.—Will you give me the calculations showing how much a 2- by 1½- by ⅛-inch cold-rolled 0.15 carbon steel angle will deflect when a load of 300 pounds is suspended from it in the manner shown in Fig. 1? We have been able to get figures for hot-rolled sections, but cannot seem to find consistent figures for cold-rolled angles.

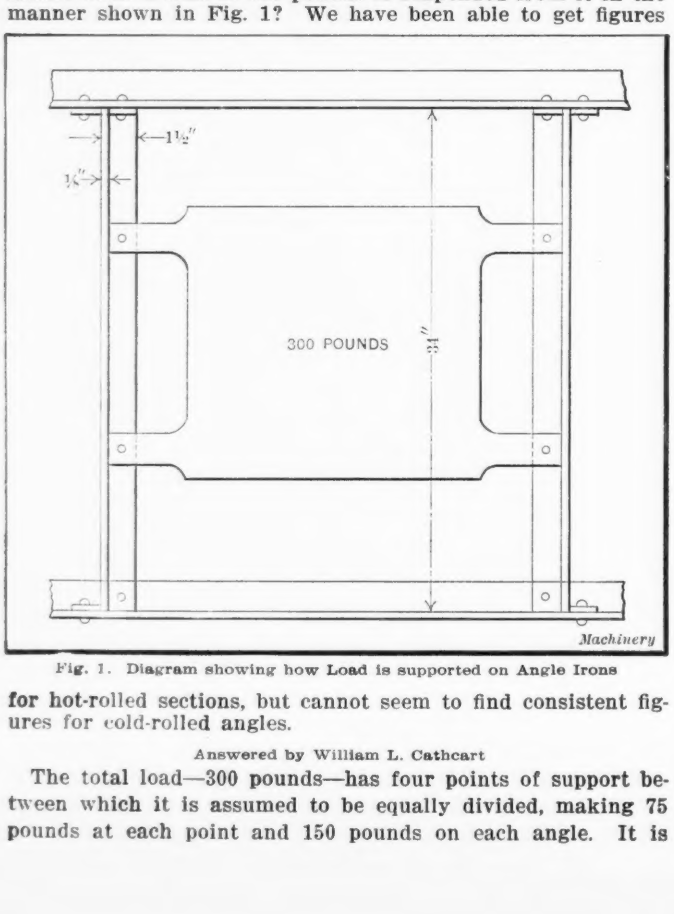


Fig. 1. Diagram showing how Load is supported on Angle Irons

Answered by William L. Cathcart

The total load—300 pounds—has four points of support between which it is assumed to be equally divided, making 75 pounds at each point and 150 pounds on each angle. It is further assumed that these points are all at the same distance a, Fig. 2, from the middle of the unsupported part of the angle. Allowing one inch for the left support, each angle may be considered as a "restrained beam" (one with fixed ends) of length $l = 34 - 1 = 33$ inches, sustaining two equal loads $P = 75$ pounds, each at a distance a from the middle of the beam. The left-hand load may, for convenience, be taken as distant $b = k \times l$ from the left support, k being the fraction $\frac{b}{l}$. Since the loads are symmetrically set, the maximum deflection y will be at the middle of the beam and will be double that caused by one load P. If, therefore, we find the deflection at the middle of the beam due to either of the loads P, and multiply it by 2, we shall have the maximum deflection y. For the left-hand load, the deflection at the middle is

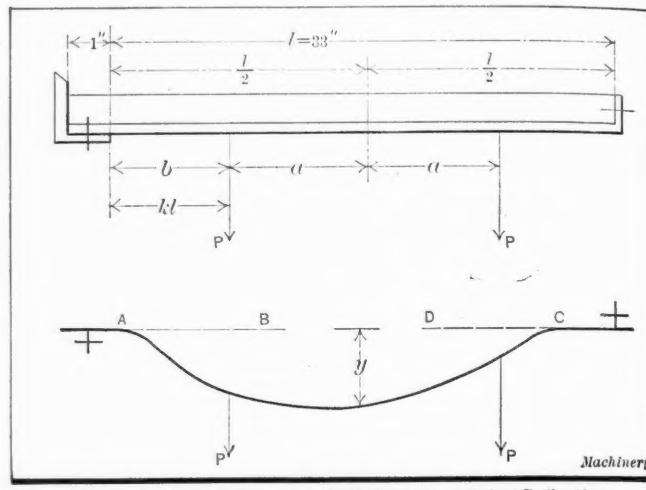


Fig. 2. Diagram illustrating Method of determining Deflection of Angle Irons

$$y_1 = -\frac{Pl^3}{EI}\left(\frac{4k^3 - 3k^2}{48}\right)$$

in which E is the coefficient of elasticity of the metal and I the moment of inertia of the cross-section of the angle about an axis passing through the center of gravity of that section and parallel to its shorter leg. For this angle the value of I is 0.17. There will be no material error, if E be given its general value for steel, *viz.*, 30,000,000 pounds per square inch, since the percentage of carbon and the physical treatment of the metal have very little effect on the value of E, as compared with their effect on the ultimate strength, the percentage of elongation, and the reduction of area. The positions of the points of application of the loads P are not given. Let these loads be 5½ inches from the middle of the angle or 11 inches from the left support. Then $b = 11$ inches, and $k = b/l = 11/33 = 1/3$. Substituting the values of P, E, I and k in the equation given above, we have:

$$y_1 = -\frac{75(33)^3}{30{,}000{,}000 \times 0.17}\left(\frac{4(\frac{1}{3})^3 - 3(\frac{1}{3})^2}{48}\right) = 0.002 \text{ inch.}$$

where y_1 is the deflection at the middle of the angle due to one load P. The total deflection at the middle, caused by both loads, is twice this amount, or 0.004 inch.

In this solution it has been assumed that the loaded angle is a restrained beam, that is, that the tangents AB and CD to the elastic curve at the supports of the angle are horizontal, which is not strictly true. Further, the accurate value of E for this case would probably lie between 29,000,000 and 30,000,000 pounds. The results are, however, sufficiently accurate for all practical purposes.

* * *

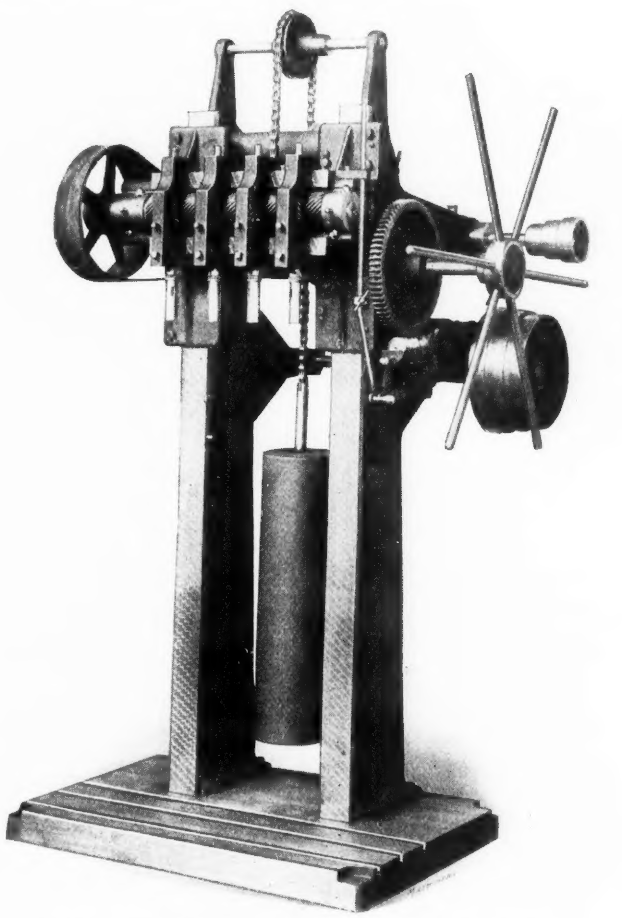
Regular machine taps often work badly when used for tapping brass. The teeth lack support and are likely to "hog in," catch and break. Two-flute taps are recommended for tapping tough brass. The lands of two-flute taps are broad and the flutes narrow, thus providing plenty of support for the teeth and preventing "hogging in." Very smooth threads can be cut with such taps if properly made.

NEW MACHINERY AND TOOLS

THE COMPLETE MONTHLY RECORD OF NEW DESIGNS AND IMPROVEMENTS IN AMERICAN METAL-WORKING MACHINERY AND TOOLS

MOLINE MULTIPLE SPINDLE DRILL

The illustration presented herewith shows a new type of multiple spindle drill which has been placed on the market by the Moline Tool Co., Moline, Ill. It will be seen that this machine is driven by the regular arrangement of spiral gears which has been adopted by this company, and that it is built



Machine built with Extended Base on which Work is mounted

without a table, the idea being to drill work set on the extended base.

The rail is of the hollow box form provided with long bearing surfaces and is securely gibbed to the columns; it is counterweighted and provided with two ways on which the heads are mounted. The main spiral gear is cut from high carbon steel; it has a helix angle of 45 degrees and is 3½ inches outside diameter. This gear is supported for its entire length in an accurately machined groove in the rail and runs in cast-iron bushings; the end thrust is taken on ball bearings, and the gear is driven by a belt provided with an idler to adjust its tension. This gear is furnished in three different pitches: a 16-pitch gear is furnished when it is desired to use the Nos. 7 and 8 heads for very light drilling or wood boring at ⅞ or 1 inch centers; a 10-pitch gear is supplied where medium drilling is to be done and Nos. 10 and 12 heads are used at centers of 1¼ or 1½ inch; and a 5-pitch spiral gear is regularly furnished unless otherwise specified, this gear being used for regular lines of work where the closest center distance is not less than 2 inches, and heads from No. 16 up are used. The heads are accurately fitted to the ways on the rail and each head is secured by two clamps. The heads may be readily removed or set at any desired point by simply loosening the clamps.

The machine shown in the illustration is built without a table and is intended for handling high work set up on the extended base. The same machine is also built with vertical T-slotted faceplates secured to the face of the columns, or with a regular table mounted on the knees and provided with vertical adjustment by means of elevating screws. When the regular table is furnished with an oil groove, it has a working face five inches wide and two T-slots. When a plain table is used it is twelve inches wide and may be furnished either with or without T-slots.

The columns have wide square ways on which the rail slides, and are securely bolted and doweled to the base of the machine to secure perfect alignment. The base is wide and heavily ribbed to prevent it from springing, and when the machine is built without a table or with vertical faceplates, the surface of the base is finished, extended forward from the face of the columns and provided with T-slots. When the machine is equipped with a table, the base is made in the form of an oil pan with a raised rib extending entirely around the outer edge.

The feed is transmitted by a rack and pinion driven by powerful worm-gears. The racks are cut from cold-drawn steel and mesh with pinions cut directly in the feed shaft. A pilot wheel is provided to operate the quick return, and the machine is also equipped with automatic stops. Three changes of feed are provided by cone pulleys which are made in sizes to provide any feed within the range of the machine.

BROWN & SHARPE NO. 00G AUTOMATIC SCREW MACHINE

The customary method of driving single spindle automatic screw machines is to employ two or more belts from an overhead countershaft: one for the feed mechanism, and one or two for the spindle drive, with sometimes an extra belt for

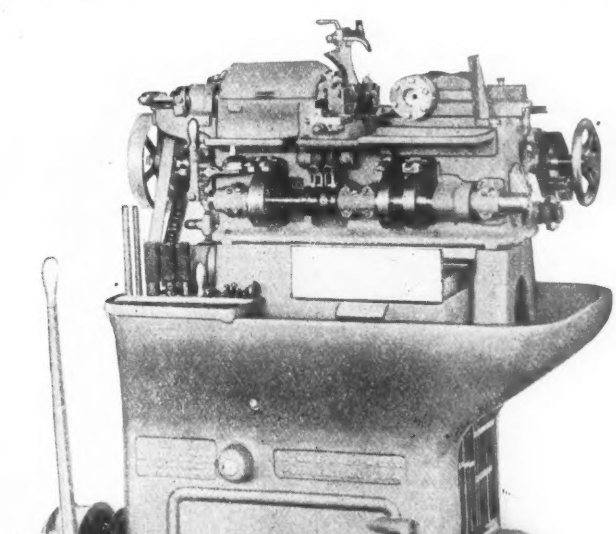


Fig. 1 The Brown & Sharpe No. 00G Automatic Screw Machine with Constant Speed Drive

the oil pump. The overhead works usually consist of two countershafts, one constant-speed driven from the main line-shaft, and one variable-speed driven from the constant-speed shaft by a belt on suitable cone pulleys. The feed mechanism is belted to the constant-speed countershaft and the spindle to

the variable-speed shaft, all spindle speed changes being made by shifting the short connecting belt on the cone pulleys. Such a system of overhead works for each machine requires a large amount of space on the ceiling, and leads to considerable expense in the installation and maintenance of a number of loose pulleys, hangers and belting. Furthermore, the shifting of the cone pulley belt for speed changes is an unhandy operation, owing to the shortness of the belt, and the manner in which it has to be done.

A recent departure in screw machine design, with a view to eliminating these difficulties, has been made by the Brown & Sharpe Mfg. Co., of Providence, R. I., in the design of their No. 00G automatic screw machine. This machine, as shown in Fig. 1, is of the constant-speed drive type; it will handle bar stock up to 5/16 inch in diameter, and turn any length up to 1¼ inch. The pulley is located at the extreme left of the machine, and the speed changing mechanism is enclosed within the base of the machine. As the driving pulley is equipped with a friction clutch, the machine can be driven from either the main line or a jack shaft, as preferred. It is, in addition, particularly well adapted for the application of a motor drive, any standard motor being employed, with the full efficiency available at all times.

Fig. 2 is a section through the base, parallel to the floor, showing the arrangement of the main driving shaft and speed change gears. A vertical section through the base is illustrated in Fig. 3, showing the arrangement of the spindle driving belts and pulleys. The pulley *A*, which is bronze bushed and runs loosely on the shaft *C*, transmits the power to the machine. The friction clutch located within this pulley is operated by the long lever shown at the left end of the machine in Fig. 1. This clutch engages or disengages the pulley *A* with the sleeve *B*, which is keyed to the shaft *C*, thus starting or stopping the machine. An important feature is the provision of an adjustable screw in the side of the starting lever, so that when the lever is moved to disengage the clutch, the screw slides upon the swinging brake shoe *T* which is faced with leather, and bears against the tight pulley *R*, thus serving as a brake to stop the spindle quickly, and a lock for the mechanism when removing the speed change gears. None of the pulleys or gears are fast to shaft *C*, except the gear *N*, meshing with gear *M*, which, in turn, drives the shaft *L* and the two gears *J* and *K* keyed to it. The pulleys *E* and *F*, and also the clutch *P*, are keyed to a sleeve *W* on which the gears *G* and *H* run free, the whole sleeve being loose on the shaft *C*. Thus, engaging the clutch *P* with the smaller gear *H* drives the pulleys at high speed, while shifting to drive by the larger gear *G* gives a slow speed in the same direction. The pulleys *E* and *F*, as shown in Fig. 3, are connected to the spindle by an open and a crossed belt, respectively. *X* and *Y* are idler pulleys which serve to guide the belts and act as tighteners, adjustments being made by sliding the round shanks of the pulley brackets backward or forward through their bearings in the bed, and clamping them in position with set-screws. *D* is a guide pulley.

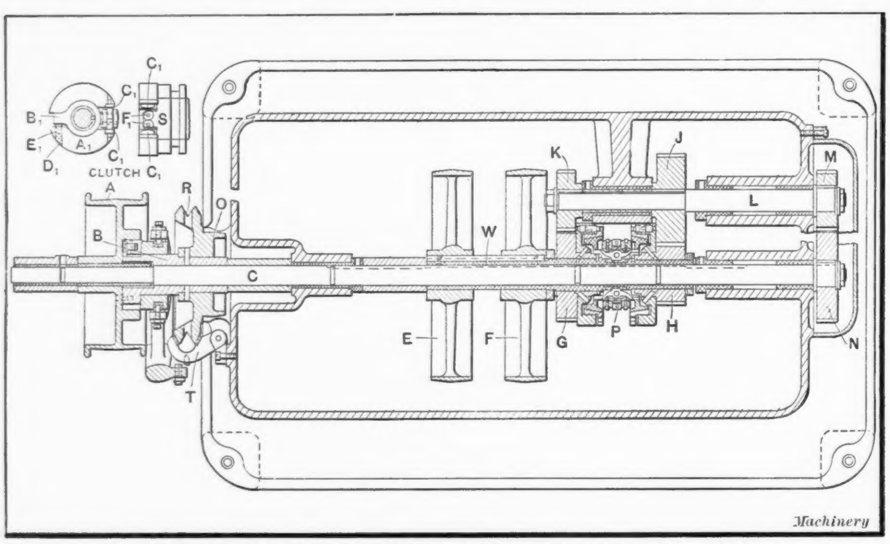


Fig. 2. Horizontal Section through Base of Machine showing Driving Shaft and Speed Change Gears

The spindle pulleys are free on the spindle, and are engaged or disengaged by a clutch similar in design to *P*, Fig. 2. Throwing in this clutch reverses the spindle, while the operation of the clutch *P* changes the speed from fast to slow, or *vice versa*. This combination of automatic spindle reverse and automatic speed changes is especially advantageous in threading, drilling with small drills, or turning work having large and small diameters, where the best results are obtained by using a slower or faster speed on certain operations than on the remainder of the work. The variations of spindle speeds are obtained by removing the two gears *M* and *N*, Fig. 2, which are held in place by nuts, and substituting pairs of different ratios. Being on the end of the base, they are readily accessible, and are completely shielded by a hinged cover when running. A large door is also provided in the base, allowing easy access to the driving mechanism. The placing of the spindle drive belts in the base of the machine allows the spindle drive pulleys and clutch to be completely shielded by a hinged cover, enabling the operator to work from any position around the machine in adjusting tools or inspecting work with comparative safety. Also, as the belts are enclosed in a guard within the base, they are thoroughly protected from the bad effects of the cutting lubricant. The cabinet type base, which contains the entire spindle driving mechanism, adds greatly to the rigidity of the machine, reducing to a minimum any vibration which might be caused by the rapidly revolving spindle.

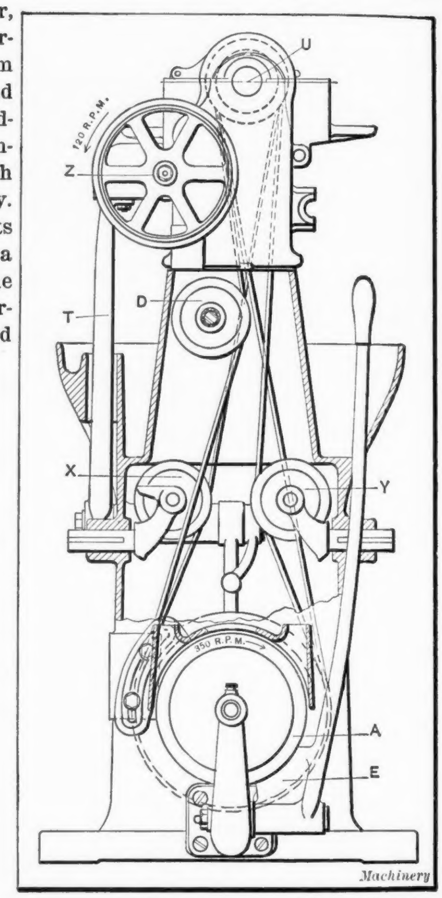


Fig. 3. Vertical Section through Base of Machine showing Spindle Driving Belts and Pulleys

The feeds and movements of the machine, other than the revolution of the spindle, are driven from the constant speed shaft *Z* at the back of the machine. This shaft is rotated by a belt from the pulley *O*, and may be stopped by the small lever at the left-hand side of the machine near the spindle, which disengages a positive clutch between the shaft and pulley, thereby stopping the entire feed mechanism. The front or timing shaft is driven from the constant-speed shaft in the rear through a worm and worm-wheel and change gears, which positively regulate the duration of the cycle of operations. The timing shaft carries the cams feeding the cross-slides, and also three carriers on which adjustable dogs are mounted. These dogs trip levers extending under the bed, causing them to engage positive clutches on the constant-speed shaft at the rear, which, in turn, drive gearing or cams for performing the various operations. The clutches automatically release after either a whole or half revolution.

The clutch *P* in Fig. 2, controlling the spindle speed change, is automatically shifted by a lever at the back of the base,

which is thrown backward and forward by a cam on the feed shaft, revolved by a clutch as previously described. The first half revolution of this cam clutch throws the lever for one speed, and the second half revolution shifts the lever for the other speed.

The construction of the friction clutch in the main driving pulley, which is also shown in detail in Fig. 2, is of an entirely different design from those usually employed in constant-speed drive pulleys. The split friction ring A_1 is expanded within the driving pulley by means of two hardened rollers F_1, carried in the outside sliding sleeve S. The two rollers bear against the hardened shoes C_1 of the friction ring, the bearing surfaces of the shoes being arcs of circles. The inside sleeve B is keyed to the drive shaft and is connected to the split ring by the extension of a section of the sleeve into a recess in the ring, shown at B_1. As the sleeve S is moved toward the friction ring by means of the fork on the operating lever, the rollers are forced between the shoes, thus expanding the ring sections and clamping the pulley to the shaft. As the sliding sleeve transmits no power, it is free from all tendency to bind. When in position the sleeve is locked, since the rollers lie beyond the center line of the arcs. Also the rolling contact, while giving a powerful expansion, requires very little effort to operate it. The amount of friction is adjusted by the screw D_1 held in place by the setscrew E_1.

As may be seen from the sectional views, the entire mechanism has been carefully designed with ample proportions to withstand the most severe service to which a machine of this size and capacity should be subjected. All bearings in the spindle drive are bushed with phosphor-bronze, and wherever it has been considered advisable, the shafts are hardened. The main drive belt is provided with an adjustable guard over the pulley, there are hinged covers over the feed shaft on the back of the machine and over the spindle drive on the top, and the operator has been protected from injury and the machine from dirt and chips in all possible ways.

NEWTON RACK-DRIVEN SLOTTING MACHINE

The illustrations show a 32-inch rack-driven slotting machine which has recently been placed on the market by the Newton Machine Tool Works, Inc., Philadelphia, Pa. The cutter bar carries a relief tool apron with vertical and horizontal steel faced clamping surfaces; this bar is counter-

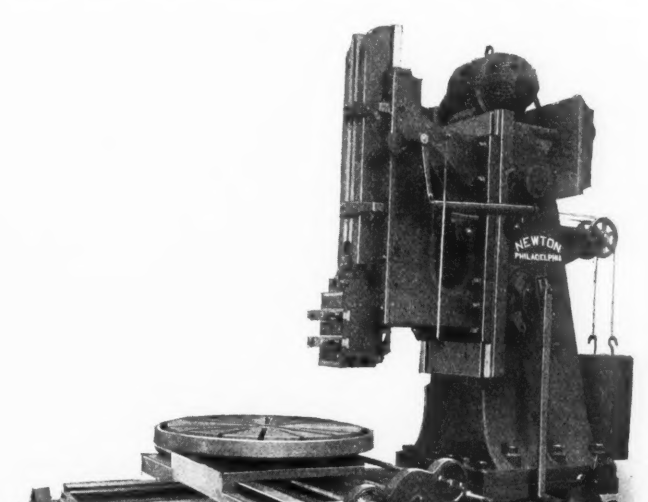


Fig. 1. Newton Rack-driven Slotting Machine

weighted and driven by a coarse-pitch steel rack. Referring to the illustrations, it will be seen that two dogs are mounted on the cutter bar, these dogs being adjustable along the bar to control the length of the stroke. When these dogs come into contact with the tripping pawl, which is connected to the master switch, they reverse the drive. It will also be seen that a lever is provided for controlling the drive by hand. The cutter bar has square lock bearings in the adjustable support, which, in turn, is bolted to the upright; and the drive is transmitted from the motor pinion through intermediate gearing to the bull-gear which connects with the cutter bar rack. The feed is taken from this gearing and carried through the machine to the patented reverse box which is arranged to give a variable stroke to the pawl controlling the reversing, circular, in-and-out, and cross feeds.

The circular table is surrounded by an oil pan which is cast solid, and the exterior of this pan is graduated into 360 degrees. The circular table is held to the cross saddle by means

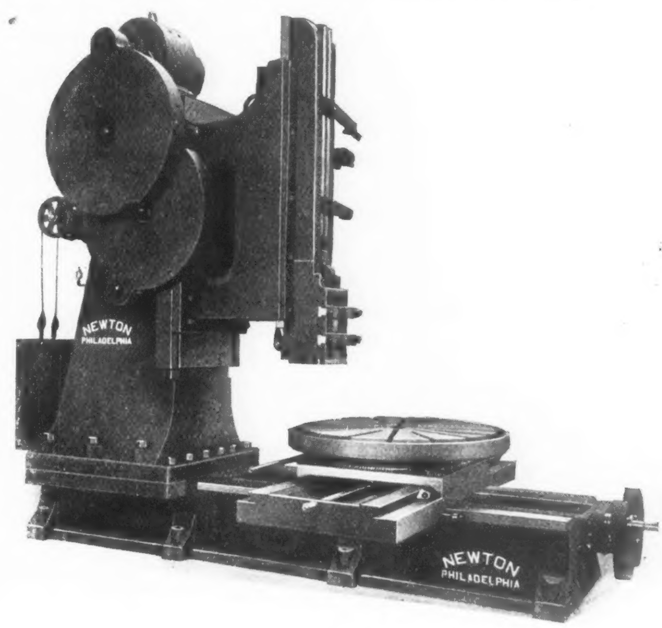


Fig. 2. View of Slotting Machine from Opposite Side

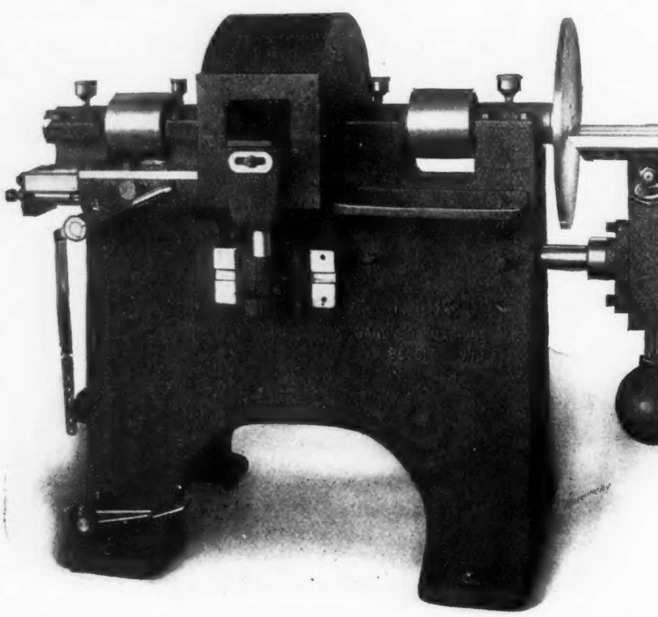
of corner clamps. The cross-slide has gibbed bearings on the base and all adjustments are made by means of taper shoes. Any of the table movements can be controlled by hand, and the motions are independently controlled by removable feed gears. By referring to the illustrations, it will be seen that particular care has been taken to protect all exposed parts of the mechanism so that the operation of the machine is made as safe as possible. The maximum length of the stroke is 33 inches, the length of the in-and-out feed is 40 inches, the length of the cross feed is 40 inches and the approximate weight of the machine is 46,000 pounds.

GARDNER DISK GRINDER

The accompanying illustration shows one of the latest machines which has been brought out by the Gardner Machine Co., Beloit, Wis. It is known as the No. 120 combination disk grinder and is equipped with three steel disk wheels 18 inches in diameter. The right-hand end of the machine is equipped with a rocker shaft and universal feed work table which is used in conjunction with the single disk wheel for grinding one surface at a time. The pair of disk wheels located at the center of the machine are used for grinding two parallel sides of a piece at the same time. The right-hand one of these two disk wheels is fixed and the left-hand wheel is mounted on a spindle carried in a movable head. By this means the distance between wheels can be adjusted so that parallel sides of pieces of varying widths can be finished. This movable head is mounted on a planed way on the machine base and is moved toward or away from the fixed wheel by means of a rack attached to its under side, which engages with a gear on the hand lever shaft. The forward travel of the head is regulated by a finely threaded stop-screw which is graduated to 0.001 inch, and this micrometer stop-screw can be set to enable the operator to produce accurate duplicate pieces. An adjustable back-stop regulates the backward travel of the head so that it only need be moved far enough to release the piece from between the wheels. This device is of special importance in grinding thin pieces, as too great an opening would make it possible for the work to be caught between the wheel and the

work rest. The hand lever shaft is connected to the foot lever shaft, so that either means of applying pressure to force the movable wheel against the work may be used, or both hand and foot pressure may be applied at the same time. A coil spring assists in the backward travel of the head.

The bearings are of high-grade babbitt metal bored, reamed and scraped to an accurate fit and lubricated by compression grease cups. The end thrust on the spindle in the movable head is always in one direction and is taken on hardened and ground steel thrust collars. Means of taking up any end play have been provided in such a way that should the length of the spindle be increased through heating, it will loosen instead of sticking in the bearings. This is also true of the spindle construction in the right-hand or fixed head, except that provision is made for carrying thrust in both directions. Different width work rests are provided with each machine and a cast-iron dust hood, which is hinged at the back, completely covers the pair of wheels at the center of the machine. The front opening in the hood can be varied in size by the use of sheet-metal slides. Abrasive ring wheels held in chucks can be used on this machine in place of the disk wheels, and with this equipment, a water attachment consisting of a water pump and connections is furnished. This machine is also made in the plain type, *i. e.*, it is the same machine throughout except that the single disk wheel at the right-hand side of the machine has been omitted, together with the rocker shaft and work table.



Gardner Disk Grinder for finishing Parallel Sides of a Piece at One Operation

WALCOTT & WOOD CRANK SHAPERS

The Walcott & Wood Machine Tool Co., Jackson, Mich., has placed on the market a line of crank shapers of the cone and gear-box drive types, which have many interesting features. As far as practicable, these machines are built on the unit system which gives free access to all working parts, and facilitates the removal of all driving gears and shafts without the necessity of disturbing the positions of the ram and rocker arm.

Fig. 2 shows a side view of the machine known as the 24-inch high-duty crank shaper, equipped with a cone-drive pulley having four steps. The gear-box crank shaper is identical in construction with the cone-drive type, except that the cone pulley is replaced by a quick-change gear box, driven by a single constant-speed pulley and friction clutch, as shown in Fig. 5. Fig 1 shows a front view of the 24-inch shaper, and gives a good idea of the massive and rigid construction of this machine.

Cross-rail and Table Details

The cross-rail on these machines is fastened to the column with the same type of bearings that are used on heavy-duty milling machines. This construction provides suitable compensation for wear and insures the cross-rail being clamped square with the column, making this machine especially suitable for the production of high-grade work. The cross-rail is of box design, rigid in construction, and has a long, wide bearing on the column. It is elevated by a screw which is provided with a ball thrust bearing to facilitate its operation. The vise furnished with these machines has a graduated base and can be swiveled and set to any angle. The base of the vise has a wide bearing surface, and is located close to the table, thus making it very rigid and capable of withstanding severe duty. The feeding mechanism for the table on the cross-rail is of simple design, positive in action, and can be quickly adjusted and reversed while the machine is in operation.

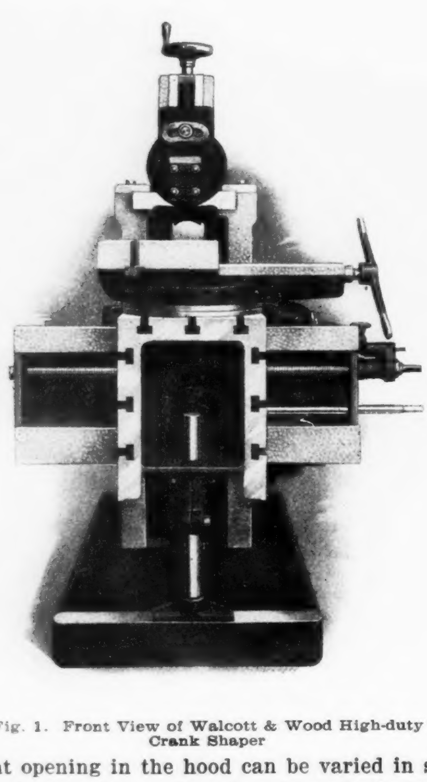


Fig. 1. Front View of Walcott & Wood High-duty Crank Shaper

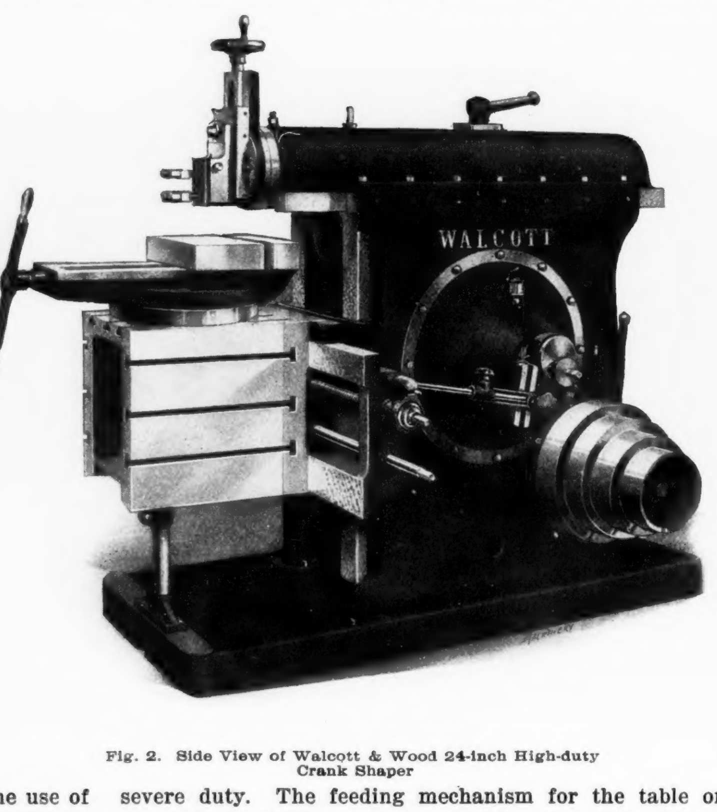


Fig. 2. Side View of Walcott & Wood 24-inch High-duty Crank Shaper

Ram, Rocker Arm and Bull Gear

The ram and rocker arm used in these machines are semi-steel castings, which greatly increases their strength and effects a corresponding reduction in weight. This permits of a greater number of cutting strokes per minute and lengthens the useful life of the machine. The bull gear pinion is made of high-carbon steel, runs loose on a high-carbon steel hardened and ground shaft, and is bronze bushed. This bushing is oiled by a large oil chamber passing through the center of the pinion shaft. The bull gear is made from a semi-steel casting, while all of the other gears used in these machines are of steel. They are of coarse pitch, wide face, and are accurately cut to insure quiet running. These features provide maximum wearing qualities and freedom from breakage under severe

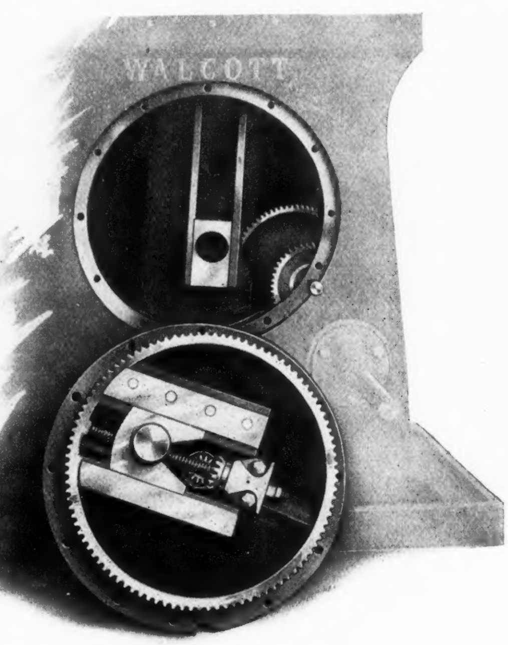


Fig. 3. Bull Gear Unit removed showing Rocker Arm and Back-gears

shock, when operating at high speed and under heavy cuts.

Fig. 3 shows the bull gear unit removed from the machine, exposing to view the rocker arm and back-gears. This illustration also shows the driving crank and its bearings which are of unusually massive construction. The housing carrying these members is accurately fitted into a recess bored in the column and is strongly bolted in position; this housing is well ribbed and is as stiff as the column itself, which is also ribbed

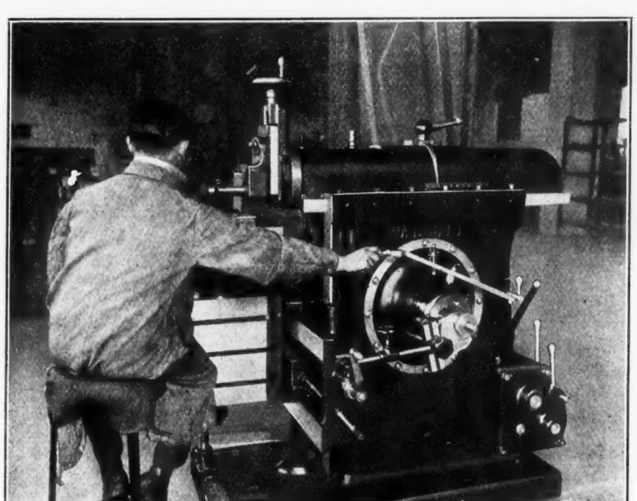


Fig. 4. Walcott & Wood High-duty Shaper equipped with Gear-box Drive

internally. Oil is supplied to the driving crank journal by a sight-feed oiler on the main bearing, which insures the proper lubrication of this important part.

Gear-box Drive and Friction Clutch

The gear-box drive type of crank shaper built by the Walcott & Wood Machine Tool Co. is illustrated in Fig. 4. All the gears and shafts in the gear box are hardened and ground and run in oil. Four speed changes are obtainable by shifting the levers on the gear box, and these speeds are multiplied by the sliding gears inside the column of the machine—as previously mentioned—thus providing for eight speeds of the ram in geometrical progression. The end view of this machine, shown in Fig. 5, illustrates the friction clutch and single pulley constant-speed drive. The clutch is of the well-known expanding brake type, and is operated by the lever which passes through the column of the machine. An extension arm attached to this lever, as shown in Fig. 4, is provided so that the operator can engage or disengage the clutch, thus stopping the operation of the machine, without moving from his position at the vise. This feature is commendable in that it is possible for the operator to stop the movement of the ram at any point of its travel.

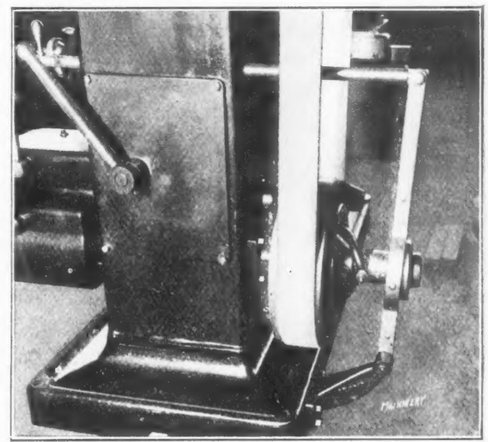
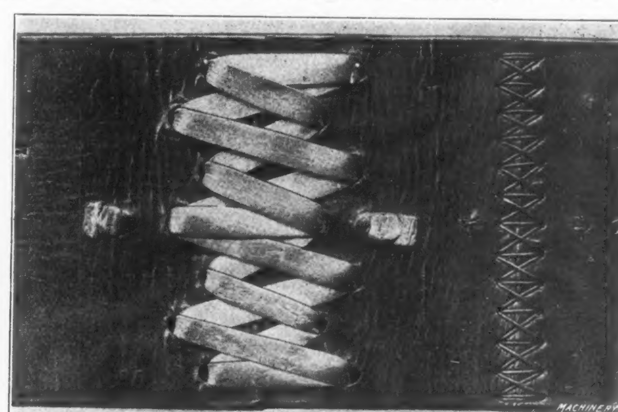


Fig. 5. Rear View of Gear-box Drive Crank Shaper showing Friction Clutch and Single Pulley Drive

Power to the tool head may be furnished, if desired, and each machine is regularly equipped with a vise, table support and all necessary wrenches. The gear-box drive type of shaper is built in 16-, 20- and 24-inch sizes.

TWISTED RAWHIDE BELT LACING

The accompanying illustration shows two methods of lacing leather belting. The first of these will be recognized as the familiar form of flat rawhide lacing which has found wide application. This method of joining the ends of a belt has two drawbacks, one of which is that the holes punched in the belt are so large that its strength is materially reduced, and



Belt laced with Twisted Rawhide Lacing and with Flat Lacing

the other that the bulkiness of the lacing at the joint has a tendency to cause the belt to jerk as the joint passes over the pulley. The other joint shown in the illustration is laced with twisted rawhide belt lacing which is being distributed in the United States by the George W. Southwick Co., 35 Warren St., New York City. This lacing consists of rawhide which is twisted into a cord, made in three sizes—14, 15 and 16 gage—and is equal in service to flat leather lacing 1/4 to 5/8 inch in width. The advantages of this method of joining a belt are: that it is only necessary to punch small holes for the lacing, so that the strength of the belt is not materially reduced; that the lacing is not bulky, so that it does not interfere with the smooth running of the belt on the pulley while the lacing is in contact with it; and that the tensile strength of the lacing is relatively high. In joining a belt with this form of lacing, it is claimed that the result approaches

a cemented splice as closely as it is possible for a laced joint to do. The tendency for the belt to jerk or jump when the lacing comes into contact with the pulley is practically eliminated and this greatly reduces the strain on the belt, so that the cost of maintenance is materially reduced.

Recent tests made at the Pratt Institute on belts laced with flat lacing and with twisted rawhide lacing showed the following results: A 3-inch single leather belt (laced by an expert) with a 3/8-inch flat rawhide belt lacing showed an opening of 1/8 inch at the abutting ends when a tensile strength of 200 pounds was reached; at 760 pounds, the opening between the ends was 1/2 inch; and when a stress of 810 pounds was reached, the belt broke at the lacing holes. A 3-inch single leather belt (taken from the same section of belting), laced with twisted rawhide lacing showed an opening of 1/16 inch at the abutting ends when a tensile stress of 900 pounds was reached; at a stress of 1460 pounds, the opening between the ends was 1/8 inch; and when a stress of 1800 pounds was reached the strands of the lacing broke. None of the lacing holes had "pulled out" or broken, and as a matter of fact the belt could have been relaced through the same set of holes. As there is no metal in this lacing, there is no danger of accidents from projecting ends.

WISCONSIN MOTOR-DRIVEN HAND MILLER

The Wisconsin Miller Mfg. Co., Milwaukee, Wis., is now building hand millers equipped with individual motor drive as shown in the accompanying illustrations. The problem of mounting an individual motor on so small a machine as a hand miller has been satisfactorily solved by providing an opening in the box shaped base at the front of the machine to receive the motor. A smaller opening is also provided in the frame at the rear to permit oiling and adjusting; this

Fig. 1. Wisconsin Hand Miller showing Chain Drive to Spindle

special box shaped base is also provided with the usual oil pan for receiving oil drippings and chips. The frame covering the opening into which the motor fits, is arranged with a shelf which extends into the base of the machine and supports the motor, and this shelf may be adjusted vertically to take up any slack or wear in the chain. The frame with the motor in place is bolted to the machine and forms a complete unit, the opening being covered with fine mesh brass screen which serves to keep out any dirt but allows a perfect circulation of air around the motor so that there is no tendency for it to heat up. A Westinghouse Electric & Mfg. Co. 1½ horsepower, direct-current, variable-speed motor, with speed control obtained through a type "H" starting and speed adjusting rheostat, is used, and connection between the motor and spindle of the miller is made with a roller chain of the

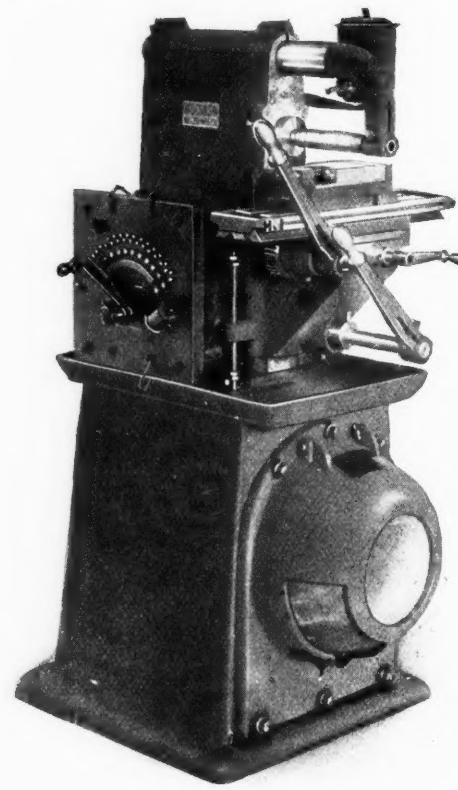


Fig. 2. Left-hand Side of Miller showing Electric Controller

type manufactured by the Diamond Chain & Mfg. Co. The spindle speeds obtainable through this arrangement vary from 260 to 780 revolutions per minute.

The spindle is made of a special steel; it is ground absolutely true and has an 11/16 inch hole through it. The taper at the front end is No. 9 B. & S. The arbor hole is ground in absolute alignment with the spindle bearings, and both the front and rear bearings are of liberal dimensions and run in self-oiling bronze boxes. The front bearing is tapered and the rear bearing cylindrical, suitable means being provided to make adjustment for wear. The arbor is made of tool steel, ground to size, and has accurately spaced arbor collars. The

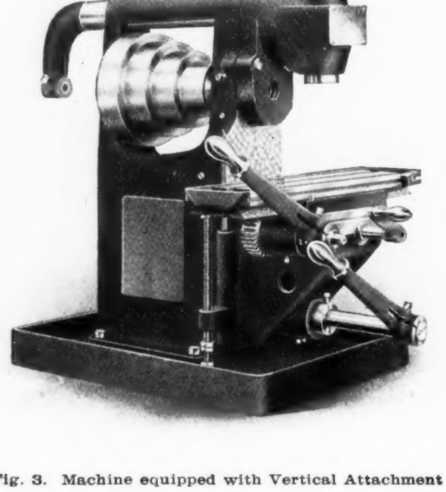


Fig. 3. Machine equipped with Vertical Attachment

pilot at the end fits a bronze bushing in the arm so that ample support is provided. The arm is of the gooseneck type and fits into a hole in the column of the machine.

The knee is of box section, has strong internal ribs, and is provided with a wide bearing on the column. The knee, saddle and table are counterbalanced by a weight inside the

column which makes vertical adjustment an easy matter. The knee can be readily clamped to the column in any position. The saddle has a large bearing surface on the knee and table, and is provided with a cross feed of 4½ inches by means of a screw and crank handle. The saddle can be clamped in any position. The table has a large working surface with oil pockets at each end, and is provided with adjustable stops for stopping the feed. A steel rack securely doweled in place below the table, provides a steady and regular table movement through either the large lever or crank. The table has a working surface of 5 5/16 inches by 17 inches. The vise is of the flanged type and has ⅝ inch slots at right-angles in the bottom; it is arranged for mounting on a swivel base. The jaws are of steel and are made removable to permit special shaped jaws to be substituted when necessary.

All of the bearing surfaces are accurately scraped and are provided with means of adjustment. The gears and shafting are all accurately machined and ground to size. The column of the machine is designed to provide for the use of a vertical attachment, as shown in Fig. 3, which has a very simple and effective drive running entirely in oil. The design is such

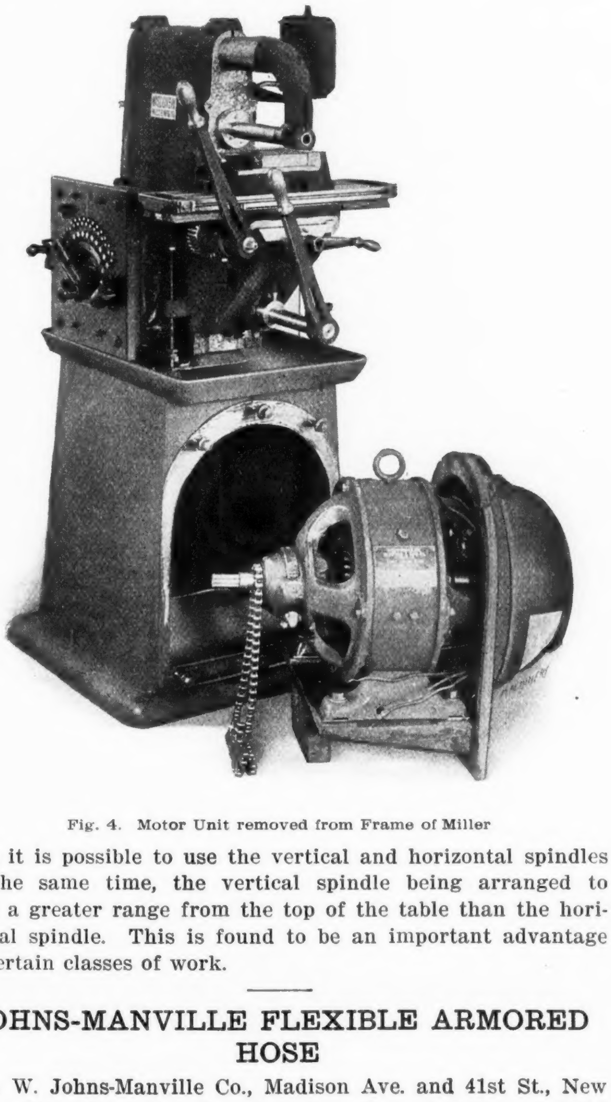


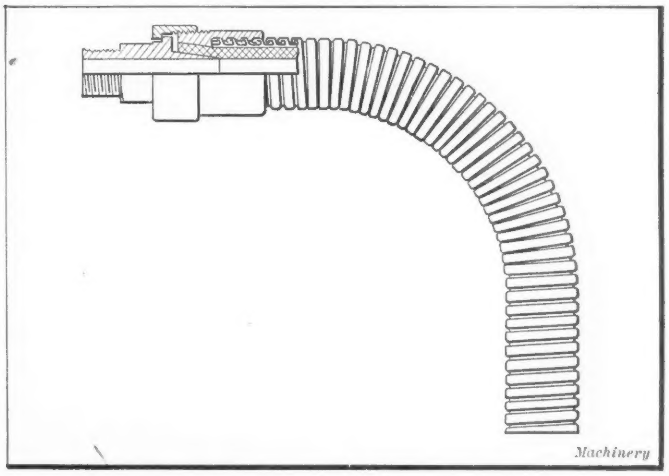
Fig. 4. Motor Unit removed from Frame of Miller

that it is possible to use the vertical and horizontal spindles at the same time, the vertical spindle being arranged to give a greater range from the top of the table than the horizontal spindle. This is found to be an important advantage in certain classes of work.

JOHNS-MANVILLE FLEXIBLE ARMORED HOSE

H. W. Johns-Manville Co., Madison Ave. and 41st St., New York City, has recently place on the market a flexible armored hose, adaptable for use on pneumatic tools, for steam connections and other classes of service. It is claimed that the design and construction of this hose is such that it will not kink, flatten, puncture or collapse. It consists of a durable rubber hose which is protected against injury from the outside by a stout metal armor. This armor is made in the form of a ribbon with crimped edges which interlock with one another and form a flexible spiral. The interlocking construction of this armor restricts curvature and thus protects the inner rubber tubing from sharp bends, kinks or from being flattened out. As a result, the hose is always wide open and the maximum flow of compressed air or steam is always passing through the connecter.

An idea of the strength of this hose in resisting pressure from the outside is obtained from tests which show that it will resist a crushing stress of from 300 to 800 pounds to each four turns of the spiral, the amount depending upon the size of the hose. The armor is also of great assistance in strengthening the hose against damage resulting from internal pressure, the armor itself being capable of withstanding the highest internal working pressures. As a result the application of this armor enables a much lighter rubber hose to be used than would ordinarily be suitable for a given class of service. As



Flexible Armored Hose suitable for use on Pneumatic Tools

the edges of the armor interlock with one another, the rubber hose is protected against stretching, all such strains being borne by the armor itself. Experiments show that a pull of from 1000 to 2000 pounds can be supported in this way.

Specially designed couplings of malleable iron or brass are furnished with each length of hose and these couplings are virtually a union of three pieces. The metal armor is riveted to an outer sleeve, which is tapered on the inside to receive the nipple and threaded on the outside to enter the lock or binding nut. This nut serves to wedge the nipple well into the inner tube and hold it securely against the walls of the sleeve. This type of hose can be furnished in any length and in any inside diameter up to 12 inches.

ATHOL VISE WITH SWIVEL BASE

The Athol Machine Co., Athol, Mass., has recently placed on the market, a new vise which combines two valuable features in its design. The first of these is the construction of the handle which connects with the head of the screw by

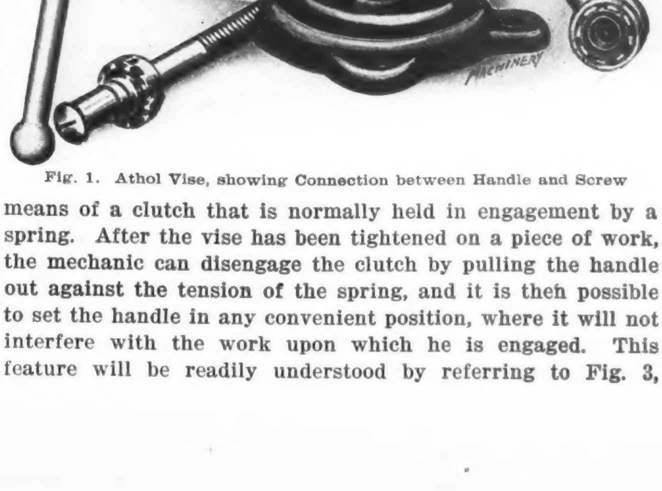


Fig. 1. Athol Vise, showing Connection between Handle and Screw

means of a clutch that is normally held in engagement by a spring. After the vise has been tightened on a piece of work, the mechanic can disengage the clutch by pulling the handle out against the tension of the spring, and it is then possible to set the handle in any convenient position, where it will not interfere with the work upon which he is engaged. This feature will be readily understood by referring to Fig. 3,

where it will be seen that the handle has been thrown down so that it does not interfere with the operator's hands when filing the piece of work clamped in the vise. Where a man is handling repetition work he can set the clutch so that the handle will come into a convenient position when the vise is tightened on the work. This means a considerable saving of

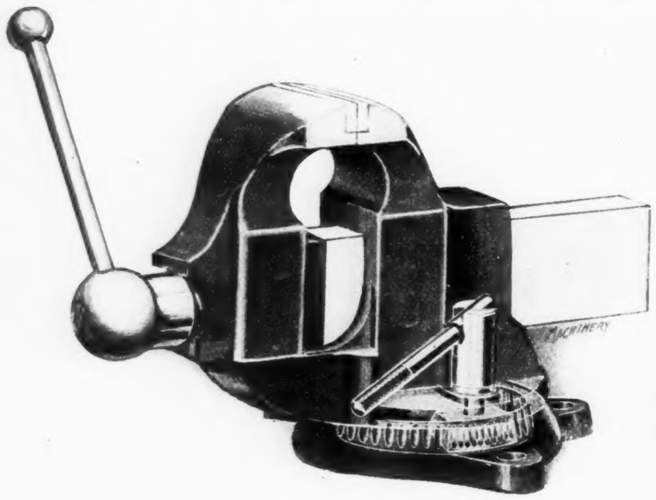


Fig. 2. Swivel Base Vise, showing Construction of Base

time, as he always has the piece set in the vise so that he can work to the best possible advantage. As the handle does not slide through the head of the screw, it cannot drop through and pinch the workman's hand.

The second feature referred to consists of the stationary base upon which the vise swivels. This base has an annular slot cored out in it, and a nut carried on the end of a bolt attached to the vise body fits in this slot. The side of the slot is laminated to mesh with corresponding laminations on the nut when the bolt is tightened. In this way a secure grip is obtained and the vise is held firmly in any desired position. This clamping nut is operated by means of the small lever shown in the illustration. A quick adjustment can be secured by set-

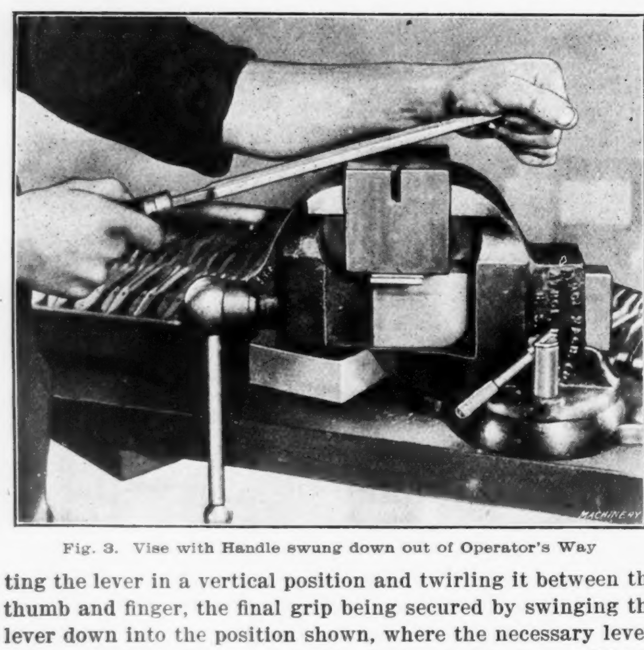


Fig. 3. Vise with Handle swung down out of Operator's Way

ting the lever in a vertical position and twirling it between the thumb and finger, the final grip being secured by swinging the lever down into the position shown, where the necessary leverage is secured. The use of this device makes it possible to set the vise in the most convenient position and then clamp it securely in place. These vises are also made with the improved style of handle previously described, but without the swivel base.

VERTICAL ATTACHMENT FOR STANDARD AUTOMATIC GEAR-CUTTERS

The Standard Mfg. Co., Bridgeport, Conn., has recently brought out a vertical attachment to be applied to any of its gear-cutting machines. The primary object of this attachment is to provide for cutting teeth on the flat sides of disks or wheels. The machine illustrated in Fig. 1 and the details of the attachment which appear in Fig. 2 show the design used for cutting the teeth on face milling cutters. The primary difference between this type of gear-cutter and the ordinary gear-cutter is that the work-holding spindle is in a vertical plane. By referring to the detail illustration, Fig. 2, in connection with the general view, Fig. 1, it will be seen that the blank in which the teeth are to be milled is shown in part at *A*. The attachment is composed mainly of three brackets, *B*, *C* and *D*, which are fastened to the under side of the bed of the gear-cutting machine. These brackets support two longitudinal shafts, one of which is shown at *E*; the

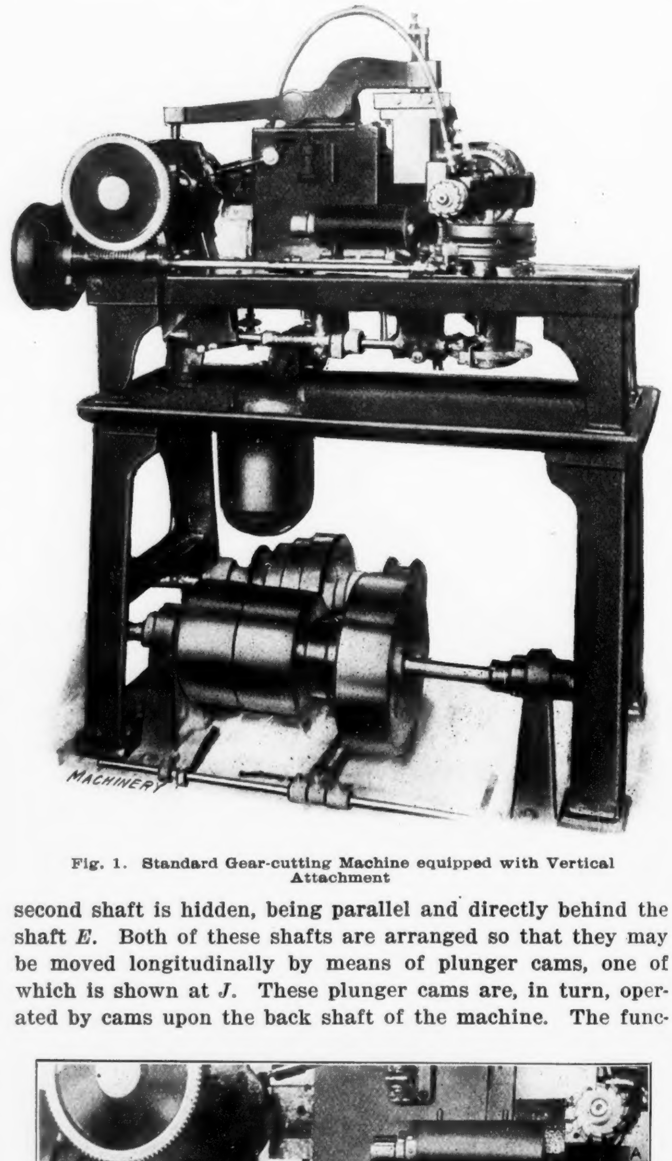


Fig. 1. Standard Gear-cutting Machine equipped with Vertical Attachment

second shaft is hidden, being parallel and directly behind the shaft *E*. Both of these shafts are arranged so that they may be moved longitudinally by means of plunger cams, one of which is shown at *J*. These plunger cams are, in turn, operated by cams upon the back shaft of the machine. The func-

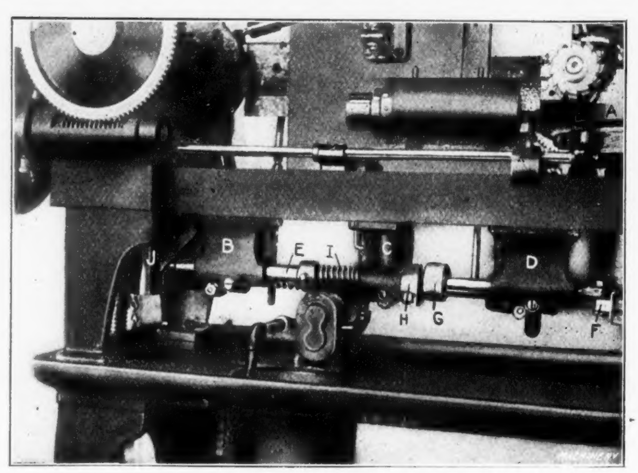


Fig. 2. Details of Vertical Attachment for Gear-cutting Machine

tion of shaft *E* is to operate the pawl *F* which engages with a feed ratchet upon the work holding shaft. A better view of this pawl is shown in Fig. 1. The second shaft, which is not visible, is so situated that a lock on its end engages the ratchet wheel on the work spindle, thus preventing the work from turning while the tooth is being cut. The two cams on

the back shaft, which operate the lock and index pawl, are adjustable in position and so timed that the plunger cam which operates the ratchet wheel lock relieves its pressure on the lock just before plunger cam J descends and forces the feed pawl against the ratchet wheel for a sufficient length of time to index the blank properly. As soon as the work is indexed, the lock drops into position in the ratchet. Means of adjustment is provided on shaft E by the nut G. To make this adjustment, the check nut H is first loosened and the nut G is then turned in the proper direction to shorten or lengthen the shaft E. A spring I serves to keep the shaft E back against the plunger cam.

The work holding arbor is provided with a faceplate, and by setting a stop, the feed may be thrown out at any desired position of the circle. There is also an index pin which may be readily engaged with the faceplate on the work arbor to lock the table while changing the work. This vertical gear-cutting attachment may be quickly removed and the machine transformed into the ordinary type of gear-cutting machine at a few moments notice.

KEMPSMITH LINCOLN MILLER

The Kempsmith Mfg. Co., Milwaukee, Wis., has recently made an addition to its line of millers by bringing out a machine of the Lincoln type designed for the manufacture of heavy duplicate parts, and several interesting features of this machine are shown in the accompanying illustrations. The machine shown in Fig. 1 is of the constant speed drive type, all changes of spindle speed being effected through change and sliding gears. For each change of spindle speed there is a corresponding change of back-gear ratio, this ratio never falling below 1 to 2.15 and running as high as 1 to 13 on the slowest speed, thus adapting the machine for taking the stiffest possible cuts with high-speed steel cutters. This feature, combined with the unusually large bearing surface of the saddle and table, and the essentially rigid construction, enables the machine to be used on a wide range of both light and heavy work.

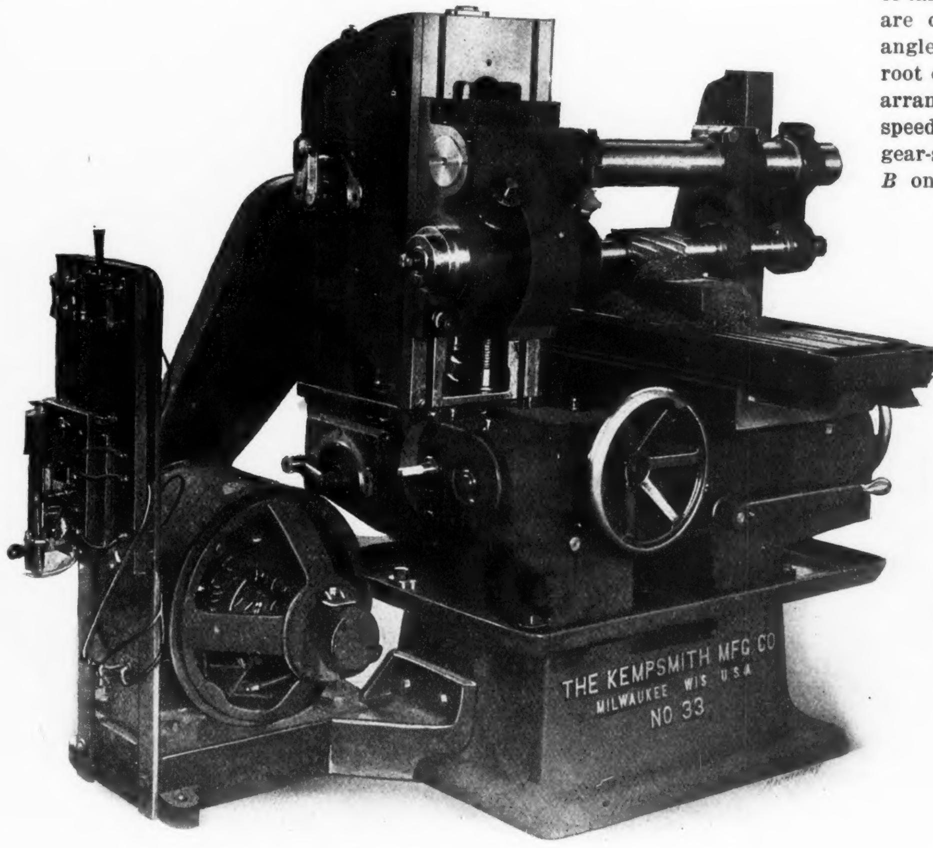


Fig. 1. Kempsmith Lincoln Miller equipped with Motor Drive

The single driving pulley, shown in Fig. 2, is 15 inches in diameter; it is flanged and carries a 4-inch double belt, the pulley running at a constant speed of 250 R. P. M. This pulley may be belted direct to the lineshaft, if desired, and this arrangement also lends itself readily to the application of motor drive as shown in Fig. 1. This driving pulley is keyed to the main driving shaft which runs in a pair of Standard roller bearings. The driving shaft is connected to the gear train through a clutch which is so designed that it starts on friction and throws in a positive three-tooth hardened steel clutch. The starting and stopping of the entire gear train and, consequently, of the spindle, is controlled by either one of two levers located at each side of the bed of the machine. To start the spindle, the operator raises the starting lever; when the lever is in its lowest position the clutch is out, so that it is impossible for jar or vibration to throw the clutch into engagement. When the clutch is engaged, the power is transmitted to the spindle driving shaft by means of a train of spur gears mounted on short, heavy shafts running in Standard roller bearings. As the changes in speed are secured entirely through sliding gears, it has been possible to make these gears of large diameter and wide face. The teeth are coarse pitch with 20-degree pressure angle, which gives great strength at the root of the teeth. With the gears A and B arranged as shown, three changes of spindle speed are obtained by shifting the sliding gear-set C. By interchanging gears A and B on their respective shafts, three other changes of spindle speed are obtained by shifting the gear-set C.

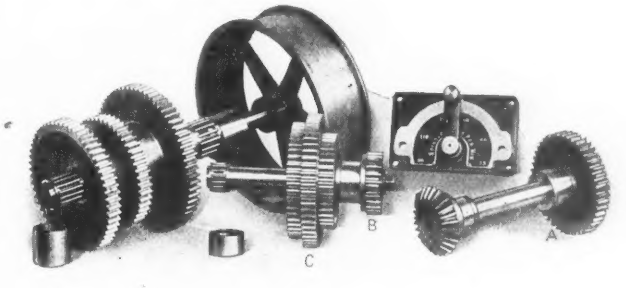


Fig. 2. Pulley, Driving Shaft and Gearing of Kempsmith Lincoln Miller

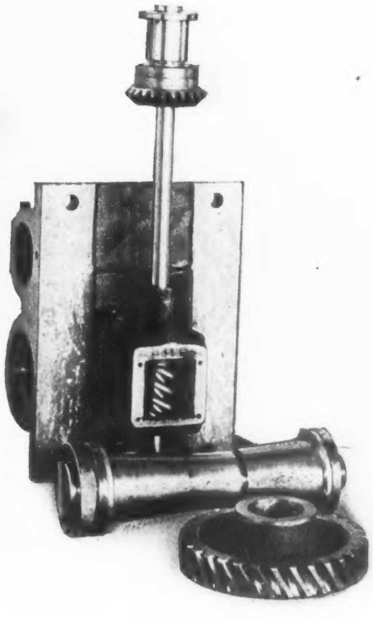


Fig. 3. Headstock showing Worm Drive to Spindle

The spindle shown in Figs. 3 and 4 is driven by means of a worm and worm-gear. The worm is of hardened steel and is carried by a vertical shaft running in roller bearings, the end thrust being taken by ball bearings. The worm-gear is made from a large bronze casting and is mounted between the spindle bearings. This construction is of considerable advantage, since it reduces the torsional strain on the spindle to a minimum. The worm and worm-gear run in a completely

enclosed case containing an oil bath for lubrication of the drive. The spindle is made from a crucible steel forging and both the bearings and inside taper hole are accurately ground. It runs in special phosphor-bronze bearings which are provided with means for making adjustment for wear without disturbing the alignment of the spindle. The principal dimensions are length 18 inches, diameter of rear bearing 2¾ inches, largest diameter of front bearing 4¼ inches, and diameter of nose 4 inches; the hole at the front end of the spindle is No. 11 B. & S. taper, and is slotted to provide positive drive for the arbor. A draw-in rod is also supplied for firmly seating the arbor in the taper hole of the spindle. The headstock in which the spindle boxes are mounted is provided with

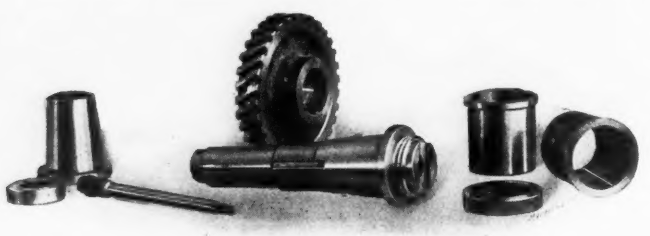


Fig. 4. Parts of the Miller Spindle

oil pockets, and sight-feed oilers show at all times the amount of oil in these pockets, thus insuring adequate lubrication of the spindle bearings.

The head frame is of massive box type construction and is braced internally to resist all possible strains. The face to which the headstock is bolted is of ample dimensions, the headstock being secured by four bolts. A heavy boss on the headstock fits accurately into a corresponding slot in the head frame, thus preserving the vertical alignment. In addition to the spindle, the headstock carries one end of the heavy overhanging arm which is 4 inches in diameter and made of solid steel, accurately ground and polished, and supported at its outer end by a substantial outboard support. This overhanging arm is furnished with an intermediate arbor support which is bronze bushed and can be used close up to the cutter. This arrangement provides positive alignment of the arbor and prevents it from being forced out of line under heavy cut. When not in use the over arm can be pushed back out of the way. Vertical adjustment of the headstock is obtained by means of a shaft projecting from the front of the bed below the head frame, instead of at the top of the head frame as in former models. This shaft is provided with a dial, graduated to thousandths of an inch, and also with a handwheel.

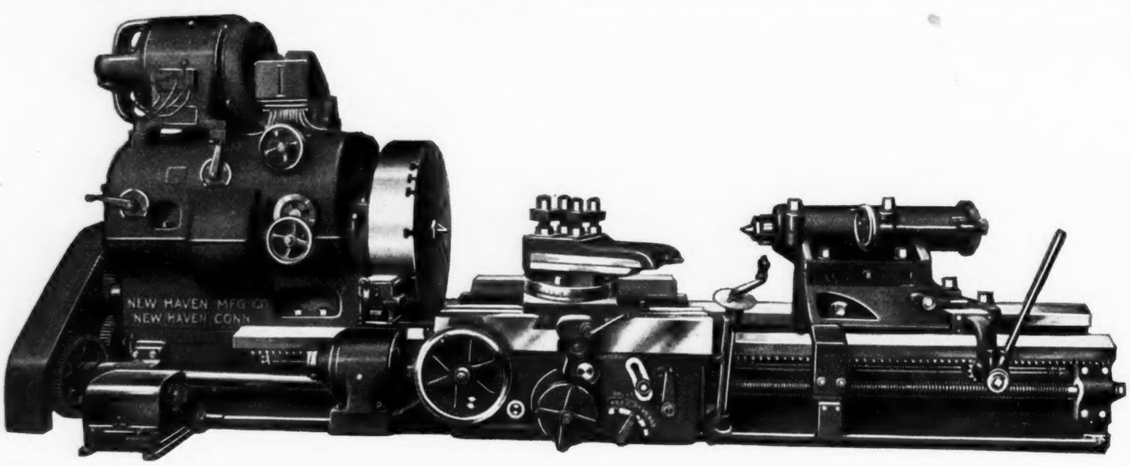
The table is only provided with power longitudinal feed, and the changes in the rate of feed are secured in the same manner as the spindle speeds are obtained. The mechanism is simple and durable, with very little possibility of getting out of order, and is totally enclosed within the bed of the machine. There are no knuckle joints and all parts are readily accessible. The power feed is controlled by means of one lever at the front end of the body, the feed at all times being engaged, released or reversed by means of this lever. The position of the lever always indicates the direction of feed. If the lever is in an upright position, the feed is not engaged, while if it is inclined to the right, the table travel will be to the right, and if it is inclined to the left the table travel will be to the left. Six changes of table feed are provided, ranging from ⅞ inch per minute to 15 inches per minute. Quick return of the table is provided through a handwheel, giving two inches of table travel for each turn of the wheel.

The massiveness and rigidity of this machine are especially noticeable in the construction of the saddle and table and in the wide bearing of the saddle upon the bed. These are all of such dimensions that a table considerably larger than the one regularly furnished can be placed on this machine without altering the saddle or bed or impairing the efficiency in any way. The table has three deep T-slots ⅝ inch in width, and is provided with ample means for returning the flow of lubricating compound. The base of the machine has a large oil pan fully protecting the floor from any dripping oil or other lubricant. To provide for the lubrication of cutters, the machine is regularly furnished with a geared oil pump running at a constant speed and driven by a Diamond roller chain. The regular equipment includes the necessary wrenches; a plain vise with jaws 7⅛ inches wide, 2 inches deep, and with a maximum capacity between the jaws of 4½ inches; and an arbor 1½ inch in diameter by 28½ inches in length from the shoulder to the nut.

NEW HAVEN FORTY-FOUR INCH LATHE

The 44-inch engine lathe shown in the accompanying illustration is a recent product of the New Haven Mfg. Co., New Haven, Conn. This machine is equipped with single pulley drive and an all-geared head. The headstock is designed to constitute a complete housing or cover for all of the gears, and it also collects all oil dripping from the bearings in the bottom of the pan; this oil may then be drawn off through a drip-cock. The headstock is 58 inches long on the bed and bolted to it with 1½ inch bolts. The spindle is made of hammered crucible steel and runs in bronze boxes equipped with sight-feed oilers. The front spindle bearing is 9 by 14 inches in size and the rear bearing 7 by 11 inches. The spindle speeds range from 0.88 to 62 revolutions per minute. Sixteen changes of spindle speeds are available with a constant speed motor.

The tailstock spindle is 7 inches in diameter by 36 inches long; it is arranged with a handwheel and gears, and has a traverse of 16 inches. Two spindle binders are provided, one at the end and one half way back on the tailstock barrel. It will be seen from the illustration that the tailstock is provided with a pawl engaging in the rack cast on the bed and a spring ratchet lever for moving the tailstock along the bed. The length of the tailstock base on the bed is 30 inches.



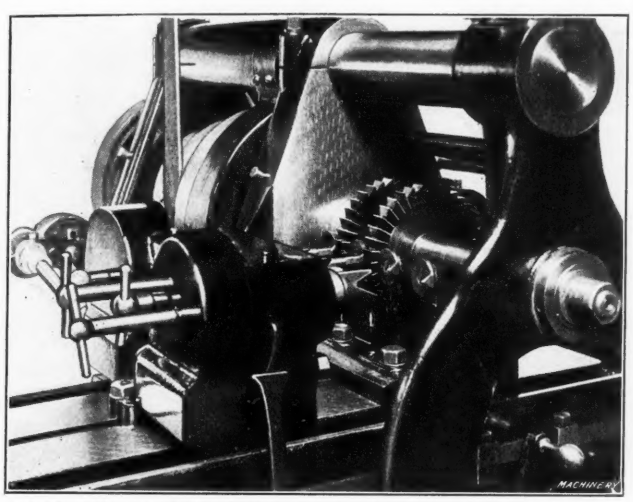
New Haven 44-inch Lathe with All-geared Head and Single Pulley Drive

The carriage bearing on the ways is 48 inches long and the width of the carriage bridge 20 inches. The compound rest has power cross, lateral and angular feeds of 20 inches. Wipers are attached to the ends of the carriage to protect the ways. The apron is of the double wall type and is cast in one piece.

All of the studs and shafts have bearings at both ends and are bronze bushed, and all of the gears in the apron are of steel. The rack pinion is made of forged steel and is 3-pitch with 3½ inch face. The reverse for all feeds is controlled from the front of the apron in a manner which makes it impossible for turning and screw cutting feeds to be engaged at the same time. If a quick traverse is required for the carriage, a small motor for this purpose can be mounted on a bracket at either end of the apron. The feeds for the carriage range from 0.010 inch to 1 inch per revolution, and the range of threads is from ½ to 16 per inch. The machine swings 46 inches over the bed and 30 inches over the carriage. With a 16-foot bed, the weight of the machine is 31,000 pounds.

BICKFORD TAP SQUARING FIXTURE FOR FLUTING MACHINE

In the New Machinery and Tools department of the February, 1912, issue of MACHINERY, the No. 3A fluting machine made by the Bickford Machine Co., Greenfield, Mass., was illustrated and described. This machine will flute taps and reamers up to two inches in diameter. The illustration below shows the application of this machine for tap squaring between centers. A special tail fixture with off-set center disks to hold the shank ends of the taps is used for this purpose, and the other ends of the work are held by special driving



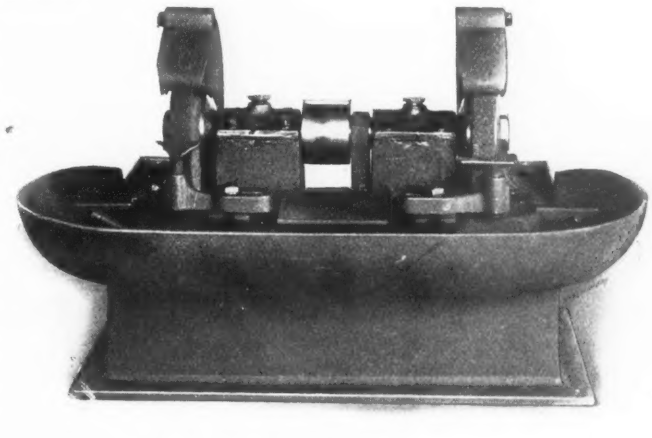
Bickford No. 3A Fluting Machine equipped with Tap Squaring Fixture

dogs having sharp corners which project inward and grip the work. These dogs are tightened up by means of the T-handle screws shown at the left-hand side of the illustration. The illustration shows only two cutters and taps in position, the other two having been removed to show the tail fixture and the method of attaching the center disks. These disks are reversible on the fixture which is arranged to be attached to the table of the machine in two positions, so that whichever way the disk is placed, the center will be in line with the head spindle. This arrangement is for the purpose of using both sides of the squaring cutters.

A special advantage of squaring taps and similar work on centers, is found in the possibility of securing absolute uniformity of corners. Only one side of the work is squared at a time, and as four pieces are being operated upon simultaneously and the return of the table of the machine and indexing is effected automatically, the operation is economical as regards its production.

DOUBLE-WHEEL STERLING TOOL GRINDER

The illustration shows a double-wheel tool grinder which has been brought out by the Sterling Emery Wheel Mfg. Co., Tiffin, Ohio. This machine is fitted with two 10 by 1 inch grinding wheels and can be used as either a wet or dry grinder according to the class of work upon which it is engaged. The grinder is of simple construction having no pump or other complicated devices to cause trouble. A water device inside the frame of the machine keeps the wheels wet when the machine is in operation, and as soon as it stops this water drains off the wheels leaving them dry and in balance at all times. This arrangement is more simple than the usual form of pump and cannot get out of order, so that there is no cost of maintenance. By throwing off the small belt at the right-hand end of the driving pulley, this water device will be stopped and the machine can then be operated as a dry grinder.


Sterling Tool Grinder equipped with Two 10- by 1-inch Wheels

This machine was designed to meet the demand for a small tool grinder for universal grinding on such work as small drills and tools. The machine is also adapted for grinding operations involved in the manufacture of small mechanical devices, such as cash registers, sewing machines, typewriters, addressing machines and similar classes of work.

GARVIN BARREL AND FACE CAM CUTTING FIXTURES

It is frequently necessary to cut cams when the quantity is not large enough to warrant buying a cam-cutting machine. To meet the requirements of such classes of work, the Garvin

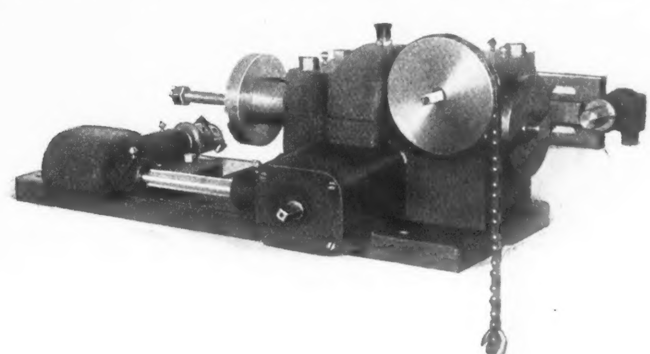


Fig. 1. Attachment for cutting Barrel Cams on a Milling Machine

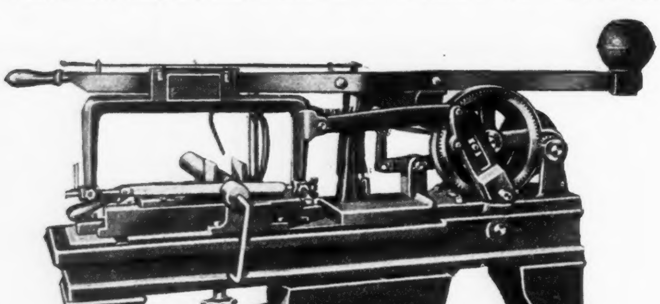
Machine Co., Spring & Varick Sts., New York City, has designed the attachments shown in the accompanying illustrations, which will fit all standard knee type milling machines. These attachments have power feed and are made reversible.

The universal joint feed rods are brought to the attachment from either the front or rear side of the milling machine frame. Fig. 1 shows the attachment for cutting barrel cams and Fig. 2 the attachment for cutting face cams. Both attachments have a capacity for cams ranging from 1½ inch to 6 inches in size. The weight of the attachments is about 225 pounds.

Fig. 2. Attachment for cutting Face Cams on a Milling Machine

WEST HAVEN UNIVERSAL HACKSAW MACHINE

The universal hacksaw machine here illustrated is a recent product of the West Haven Mfg. Co., New Haven, Conn. The special features of this machine may be briefly outlined as follows: The blade is lifted entirely free from the work during the return stroke and this quick return stroke is in the ratio of 3:1. The machine is arranged for two fixed strokes of five and seven inches. The mechanism which provides the automatic lift for the blade on the return stroke is provided with means for making adjustment for wear. All parts of the machine are made interchangeable, and, as a result, immediate shipment of any duplicate part can be obtained. Referring to the illustration, it will be seen that the machine is equipped with a pan and pump for lubricating the work and a movable vise, each jaw of which is adjustable, thus making it possible to cut angular as well as square work. The machine has a capacity for sawing work up to six inches in diameter or width and runs at a speed of 240 revolutions per minute. Twelve- or fourteen-inch saw blades are used. The machine occupies a floor space of 52 by 16 inches and has a net weight of 425 pounds.



West Haven Universal Hacksaw Machine

GLEASON TEMPERING MACHINE

The illustrations show a machine for tempering automobile bevel driving gears, which is the invention of Leon Slade and Alton Slade, and is manufactured by the Gleason Works, Rochester, N. Y. Although this machine was designed for tempering automobile bevel gears, it could also be used to advantage on a variety of work which has a tendency to warp when tempered in the usual way.

A general view of the machine is shown in Fig. 1. Briefly speaking, this machine may be said to consist of a plunger operated by compressed air. The gear to be tempered is placed on a lower die which is counterbalanced by a weight of one ton. A split bushing fits the bore of the gear, and when the plunger descends a pilot enters this bushing and expands it to keep the bore of the gear in shape. The face of the gear enters the upper die carried by the plunger and is held in shape by this means. The pilot automatically locates the gear to enter the die, and when the pressure in the air cylinder has reached the required figure, the ram forces the die and gear down into the oil reservoir and holds it there until the work has been thoroughly cooled.

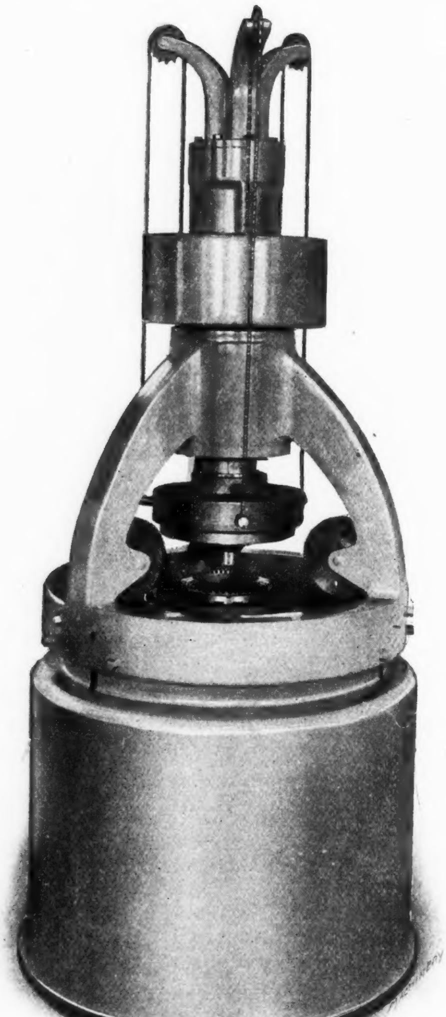


Fig. 1. Gleason Tempering Machine

Referring to the illustrations shown in Figs. 2 and 3, the operation of the machine may be outlined as follows: The work is placed in position on the lower die and when the pilot end of the plunger enters the split centering bushing it brings the bore into almost the required central position. A tapered sleeve or expander on the plunger then enters the tapered

bore in the split centering bushing and spreads it sufficiently to fit the bore of the gear when it is expanded by heat, thus providing for having the work accurately centered. The application of further pressure causes the springs above the expander to be compressed, thus applying pressure to the bore of the gear to prevent it from going "out of round." As the ram descends the upper die comes down to bear on the face of the gear and on the web, and holds the work from warping out of shape. The pressure in the air cylinder then increases until it is able to lift the circular weight which counterbalances the lower die plate. The gear is then forced down into the oil and held there until it is cold. The air pressure applied on the plunger which carries the upper die is controlled by a hand operated valve of the type used on an air hoist.

Both the upper and lower die plates are cored out and provided with grooves in their surfaces to allow a free circulation of oil. The result is that the gear is cooled quickly on all sides and comes out uniformly hard and of the same degree of hardness that it would have if dipped in an open tank. The capacity of the machine ranges from 15 of the largest sized bevel driving gears to 30 of the smallest sized gears per hour. Of course, gears of heavy section take longer to harden than lighter ones of the same diameter, but the figures mentioned are a conservative estimate of the machine's capacity.

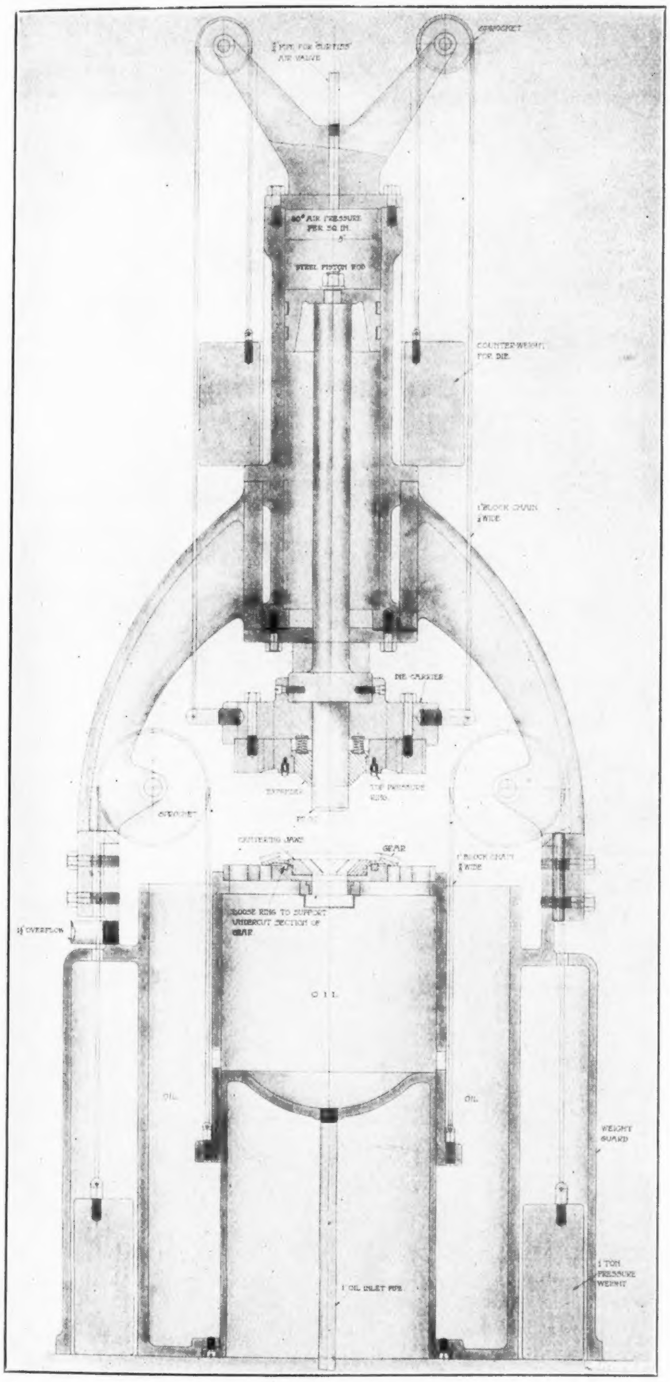


Fig. 2. Cross-Sectional View of Machine, showing Dies Open

Pipe connections are arranged to bring the oil into circulation from an outside reservoir or cooling system, and the use of a power house water heater fitted up as a radiator for cooling the oil is recommended. In the hardening room of the Gleason Works, fuel oil is circulated through the outer jacket of the radiator but water may be found more convenient for this purpose under average conditions.

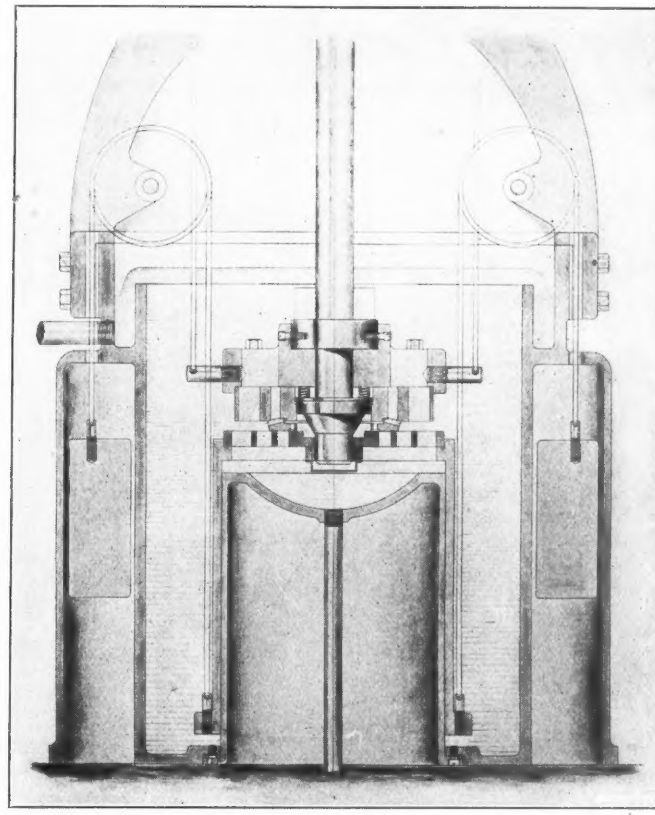
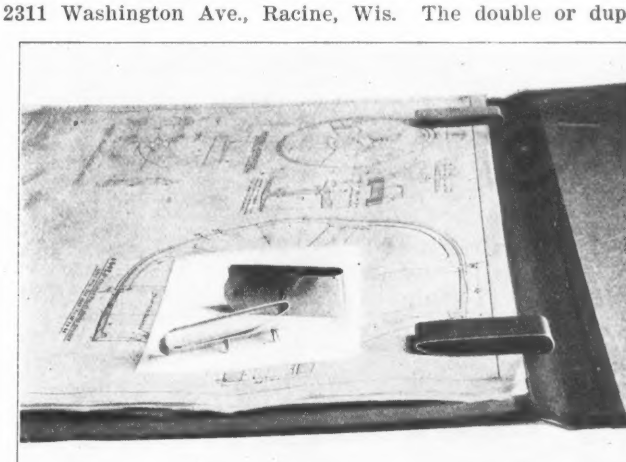


Fig. 3. Cross-Sectional View of Machine with Work Immersed in the Oil

DUPLEX SPRING CLIP FOR TRACING FILES

The accompanying illustration shows a spring clip for tracing files, blueprint files and various kinds of loose-leaf devices, which has recently been put on the market by George G. Dana, 2311 Washington Ave., Racine, Wis. The double or duplex spring of this clip applies sufficient pressure on the contents of the file to prevent it from slipping out while the portfolio is being handled. It also prevents tracings from becoming curled or folded on the upper edge, rendering them worn or creased and hard to handle when making blueprints.



Duplex Spring Clip for Tracing Files

From a glance at the illustration it will be seen that the clip is so shaped that a tracing or blueprint may be easily slipped into its place or removed from the portfolio as desired. The spring holds the outer sheets from slipping about or curling up and any sheet may be removed when desired, with-

out disturbing other sheets in the portfolio. These clips are made of polished steel and are attached to the back cover of the portfolio by clinched rivets.

BROWN & SHARPE PLAIN GRINDING MACHINES

A distinct departure from previous practice in the design of grinding machinery is illustrated in the accompanying views of two plain grinding machines which were recently

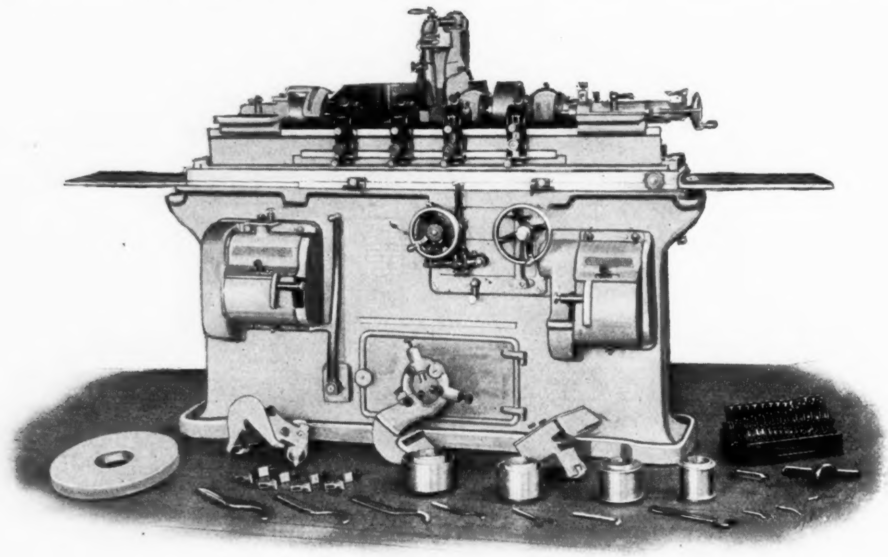


Fig. 1. No. 12 Plain Grinding Machine taking Work up to 8 inches in Diameter by 36 inches Long

built by the Brown & Sharpe Mfg. Co., of Providence, R. I. These machines are two of the largest in their line, the No. 12 taking work up to 8 inches in diameter by 36 inches long, and the No. 16 having a capacity up to 10 inches in diameter by 72 inches in length. These machines are intended primarily for manufacturing purposes and embody several new and important features in grinding machine construction, all of which increase the ease and economy of operation, resulting in greater production, without in any way impairing the accuracy of the product. The noticeable feature of these machines is the method of obtaining in the machine the speed changes for the wheel spindle and headstock, and also the feeds for the table, which, in common practice, are usually obtained by shifting belts on three sets of cone pulleys on three shafts in the overhead works. The construction has be n considerably simplified in this design by eliminating one of the shafts and all of the cone pulleys, the speed and feed changes being made by gearing in the machine.

A simple countershaft is driven from the lineshaft at constant speed, carrying two tight pulleys, one of which drives the wheel spindle and the other the driving pulley *B*, shown in the horizontal section of the machine, Fig. 3. The pulley *K* transmits power to the quick change gear box *G* on the front of the bed at the left, where the changes of speed for rotating the work are made. Thence the power is carried to the pulley *C* through the shaft *D* on the rear of the machine to the pulley *E*, and from there to the overhead drum, which carries the belt for revolving the headstock. The pulley *F*, on the right-hand end of the driving shaft *A*, which receives power from the pulley *B*, is connected to the quick change gear box *H*, where the table feed changes are made. The gearing in this box is directly connected to the front plate *M*, which carries all the mechanism for the table feed and reverse, and also for the transverse feed of the wheel slide. This plate, with all the gearing attached, can be easily removed, giving ready access to the parts. The water pump is driven from the pulley *J* fastened to the driving pulley *B*. The lever *L* on the front of the machine operates a friction clutch within the driving pulley *B*, thereby disengaging the shaft from the pulleys *B* and *J*, thus enabling the operator with one motion to stop the table and work, without stopping the water pump.

The wheel spindle is driven directly from the countershaft, which runs at constant speed. A set of split pulleys of several diameters is provided for the self-aligning wheel spindle, the different speeds being obtained by changing the pulleys, which is accomplished without removing the belt, thus giving wheel speed changes directly on the machine. The slack of belt, due to changing of the pulleys and the different positions of the wheel slide, is automatically taken up and an even

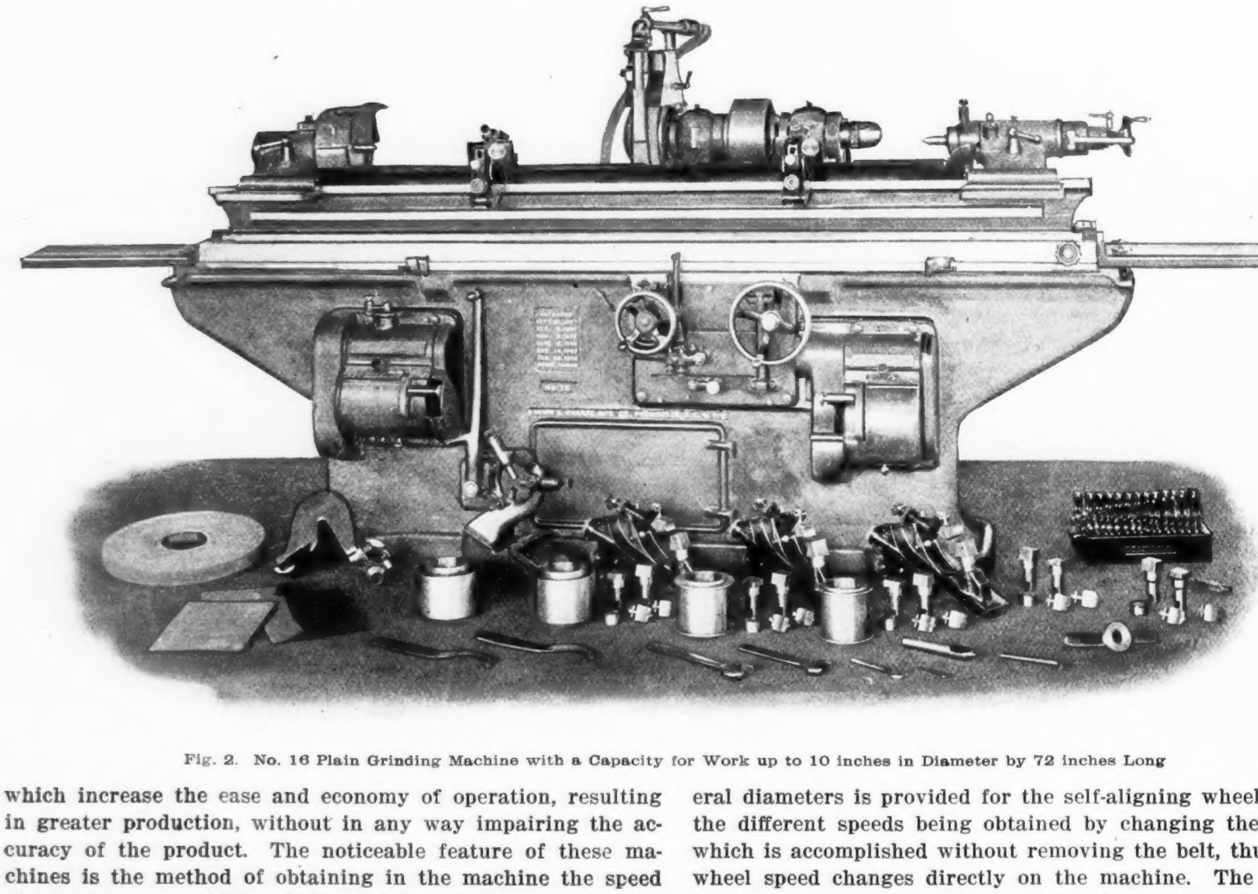


Fig. 2. No. 16 Plain Grinding Machine with a Capacity for Work up to 10 inches in Diameter by 72 inches Long

tension maintained by a spring actuated idler pulley on the countershaft.

The complete separation of the work speeds and table feeds enables the highest efficiency to be obtained at all times, as any combination of speed and feed in the entire range is instantly available, the variations being effected by gearing controlled by the simple adjustment of index slide and levers. The gear cases are located on the front of the machine and the gears are hardened. The case on the left gives the work speed changes in revolutions per minute, and the right-hand case gives table feed changes in inches per minute. Each case consists of a cone of gears, with a swinging idler arranged to mesh with a long pinion and any one of the gears in the cone with which it is firmly locked. The teeth on the idler and cone gears are of a special pointed form, so that they always drop into mesh when thrown in any position. To change the feed or speed, the locking pin of the lever on the side of the gear case is withdrawn and the lever moved down as far as possible; the index slide is next moved along until it is under the column on the index plate containing the required feed or speed. The large lever is then moved up as far as it will go, the locking pin drops into the correct hole and the proper gears are in mesh. Two series of speeds for the work are provided, one fast and one slow. These are regulated by the simple movement of the lever *X* on top of the left-hand gear case. A fast and slow table feed in the ratio of 3½:1 is instantly available in each setting for roughing and finishing work, the change being made by shifting the lever behind the table handwheel.

The table reversing mechanism is unusually rigid and accurate, and permits of work being ground close to a shoulder. The table handwheel is automatically disconnected and does not rotate when the power feed is in use. By pressing the knob in the center of this handwheel at any time during the table travel, the feed is stopped at the reversing point and the handwheel is engaged. This enables the operator to grind a shoulder on the work, by using the handwheel to adjust the table, then by pulling out the knob to again engage the automatic table feed and disconnect the handwheel. The arrangement for disengaging the transverse feed mechanism from the wheel slide by means of the small lever at the bottom of the front plate is an especially convenient feature, as it allows a quick movement of the slide by means of a handwheel for the purpose of lubrication, or adjustment when setting the wheel to the work. Provision is also made for locking the pawl in the transverse feed ratchet, so that when used with the stop-pin it insures a positive stop.

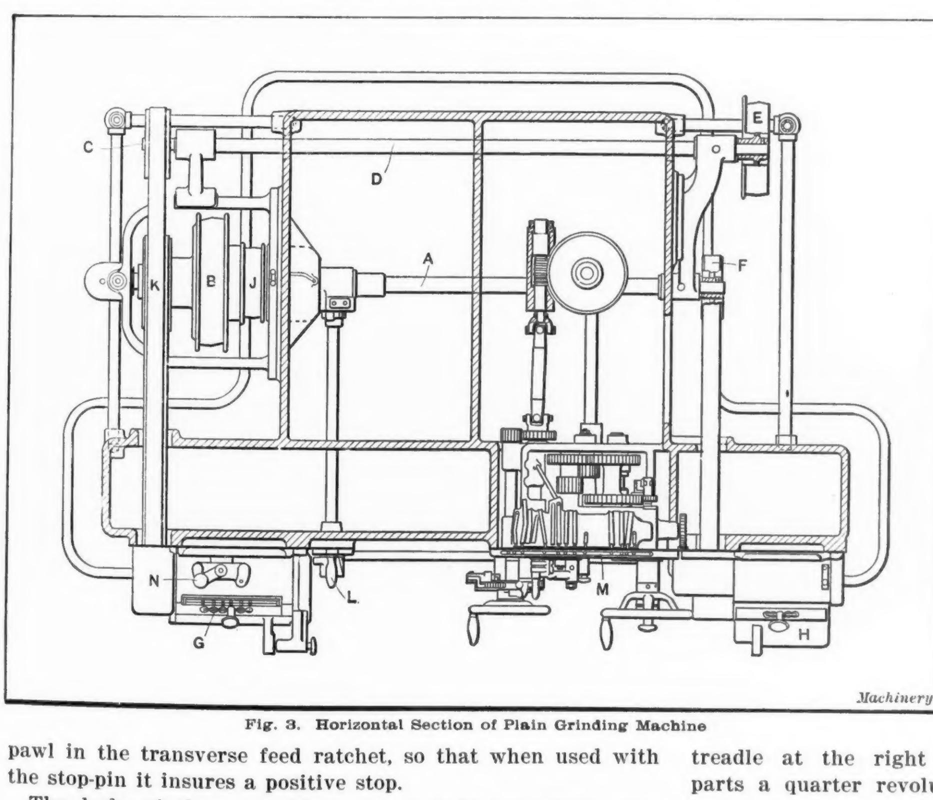


Fig. 3. Horizontal Section of Plain Grinding Machine

The beds of these machines are one-piece castings, constructed with rigid internal cross bracing and with no overhang of the ways on the smaller machine, all of which is conducive to stability, which is an important requirement in the production of accurate ground work. The base has a rim cast around it, which keeps all waste oil and water off the floor. The scale for setting the swiveling table to grind tapers is graduated in degrees, in per cent (metric) and in inches of taper per foot. Special attention is called to the concentration of control resulting from this new design. All levers and handles are within easy reach of the operator, without moving from his position.

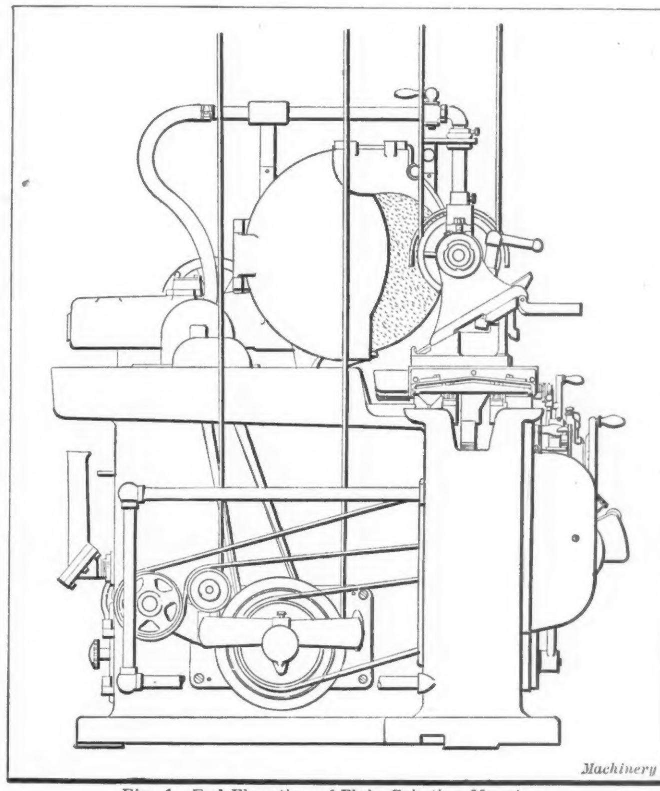


Fig. 4. End Elevation of Plain Grinding Machine

LANGELIER SHELL DRILLING MACHINE

The machine shown in Figs. 1 and 2 was recently designed by the Langelier Mfg. Co., Providence, R. I., for simultaneously drilling four holes 3/32 inch in diameter and 90 degrees apart through the walls of a brass shell. These shells are 0.410 inch in diameter by 2 1/16 inch in length, and the holes are drilled at a distance of ½ inch from the end of the shell. During the tests to which this machine was subjected before being shipped, a production of 15 shells per minute—each drilled with four holes—was obtained, which would mean an output of 9000 shells during a ten-hour day if this rate could be maintained.

The machine consists essentially of four spindles which work simultaneously; each of these spindles is located at 90 degrees from the adjacent spindle on each side. The drill spindle frames are carried by four arms of a cruciform casting, which also supports a trunnion shaft at the rear on which the four-segment edge-cam which actuates the spindle frames is mounted, together with the driving belt idler pulley brackets. The four spindles are fed to the work simultaneously by depressing the foot treadle at the right of the machine. This treadle imparts a quarter revolution to the edge-cam, which engages a roller on each drill spindle and, by so doing, draws in the four spindles simultaneously. The return of the spindles

is effected by the tension of a spring located inside each drill frame. Each of the spindles is hardened and ground and provided with two splines for balancing it at high speed. A cast-iron belt guard is provided over each spindle pulley to prevent injury to the operator in case of the belt breaking. The lower vertical spindle is completely protected from chips or dirt by means of a metal hood which is clearly shown in Fig. 1. The spindles are relieved of the pull of the belt by having the spindle bushings in the drill frame bearings extend to a position midway of the width of the pulleys, so that only the width of the driving key is left uncovered by the bushings. The hub of each spindle driving pulley runs on the projections of the bushings, thereby relieving the spindles themselves of all belt tension. This construction is virtually equivalent to having each spindle of the floating type, insuring great sensitiveness of feed and return, and complete freedom of movement at all times, even when the belt is unusually tight. Each spindle is equipped with a No. 11 Skinner chuck for taking drills up to 7/32 inch in diameter if desired. A single endless leather belt, guided by suitably placed idlers, drives all four spindles of the machine. This belt is kept at the proper tension by adjusting the large idler pulley at the left of the machine and clamping it in place by means of a single cap-screw. This arrangement will be readily understood by referring to Fig. 1. A cork insert driving pulley is used to obtain the greatest possible transmission efficiency. The pulleys on the countershaft are designed to run at 575 revolutions per minute and this provides a speed of about 2700 revolutions per minute for the spindles. Special care has been given to the provision of ample lubricating facilities for efficiently oiling all of the bearings in the machine.

The work holding bushing is mounted at the front of the cruciform casting. This bushing is of hardened and ground steel and, in addition to holding the work while it is being drilled, it acts as a guide for each drill. The shell projects out of the bushing for a sufficient distance, so that it can be readily handled by the operator.

The principle on which this machine is built might readily be modified to adapt it for a variety of classes of service. For instance, instead of having only four spindles, a greater or smaller number might be spaced around the work to drill the required number of holes simultaneously. In case it was necessary to stagger the holes, some of the spindles could be set further in or out from the frame of the machine, thus permitting some holes to be drilled at a greater distance than others from the end of the piece. Instead of single spindles, the machine might also be provided with multiple drilling heads, of the type regularly made by this company, and a large number of holes could then be drilled with each spindle. The machine shown in the illustration stands 69 inches high and the work is at a height of 56 inches from the floor. The machine occupies a floor space of 24 inches wide by 28 inches long and weighs 570 pounds. It could, of course, be built in larger or smaller sizes to meet the requirements of different classes of work.

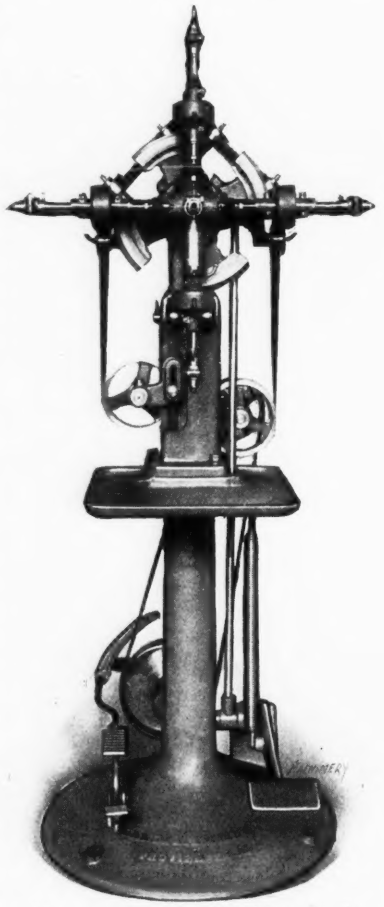


Fig. 1. Front View of Machine showing Spindles and Work Holder

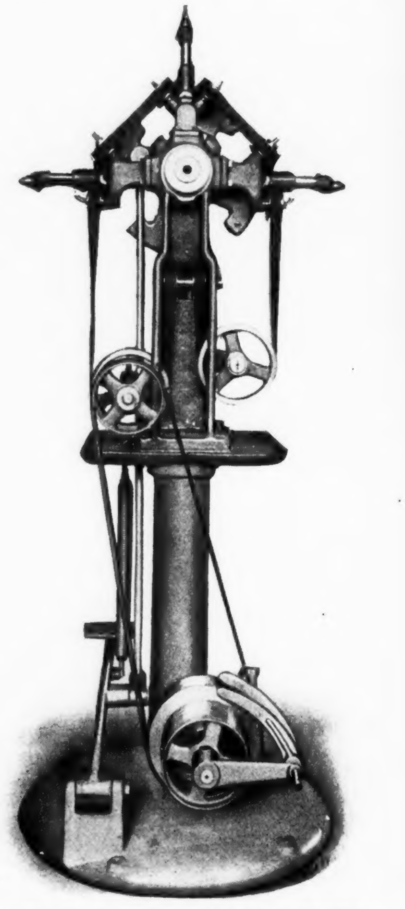


Fig. 2. Rear View of Machine showing Arrangement of the Drive

NEWTON COLD-SAWING MACHINE

A 26-inch cold-sawing machine designed for use in steel foundries by the Newton Machine Tool Works, Inc., Philadelphia, Pa., is illustrated in Figs. 1 and 2. This machine has a capacity for cutting off gates or risers up to 8½ inches in diameter at a single cut, or larger sizes can be handled by turning the work. It is equipped with one of the Tabor Mfg. Co.'s new type of high-duty blades, 28 inches in diameter, which has a tempered steel body with hardened teeth and retaining wedges. This equipment enables the machine to take feeds of ⅞ inch to 1 inch per minute.

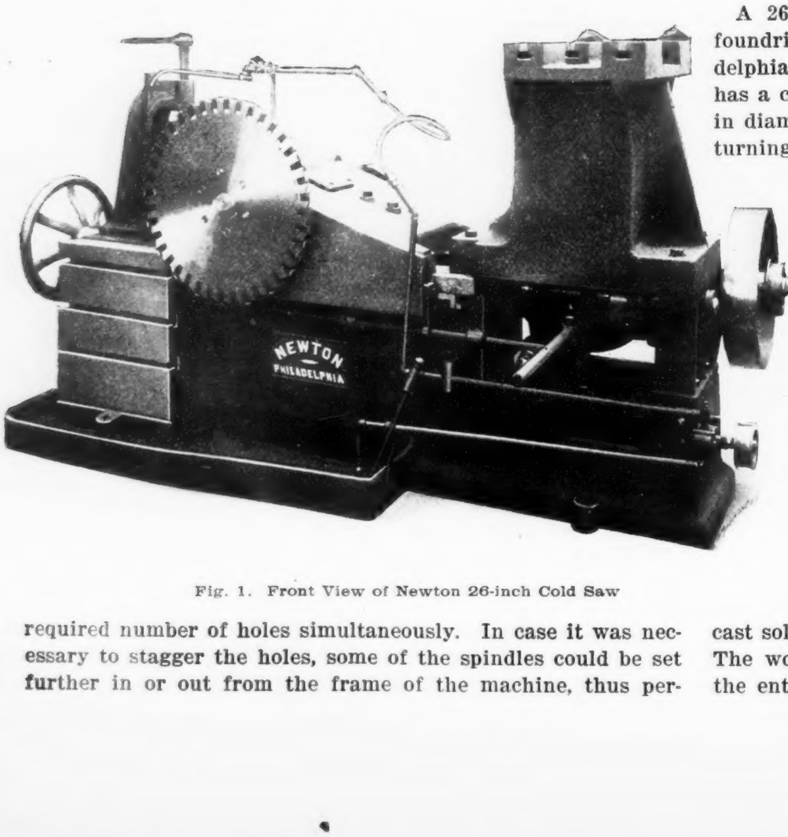


Fig. 1. Front View of Newton 26-inch Cold Saw

The design of this machine conforms with the standard Newton construction. The spindle is supported on both sides of the machine by cap-bushed bearings, and the driving spur gears are of high-grade hammered steel with the driving pinion teeth cut from the solid worm-wheel shaft. The driving worm-wheel is machined from a solid bronze casting and the driving worm is of hardened steel. The shaft supporting the driving worm has a bearing on each side of the worm and the thrust is taken on roller bearings. The saddle is equipped with lock bearings on the base with underlocking gibs cast solid, adjustment being provided by means of taper shoes. The work table has T-slots milled from the solid metal and the entire base, including the oil pan and work table, is one

solid casting. The feed-screw is supported on both ends to insure having it operate under tension at all times and the control levers are conveniently located at the front of the machine. The friction feed has a range of from ½ inch to 2 inches per minute. The machine is ordinarily driven by a 10 H. P. motor mounted on the top of the high bracket, pro-

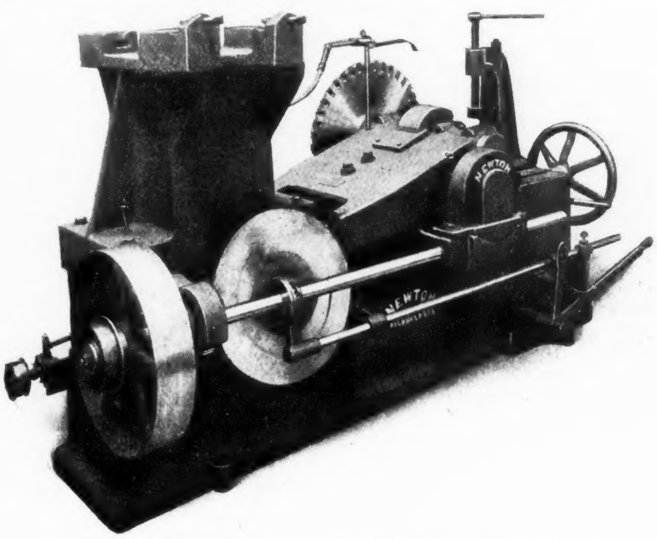
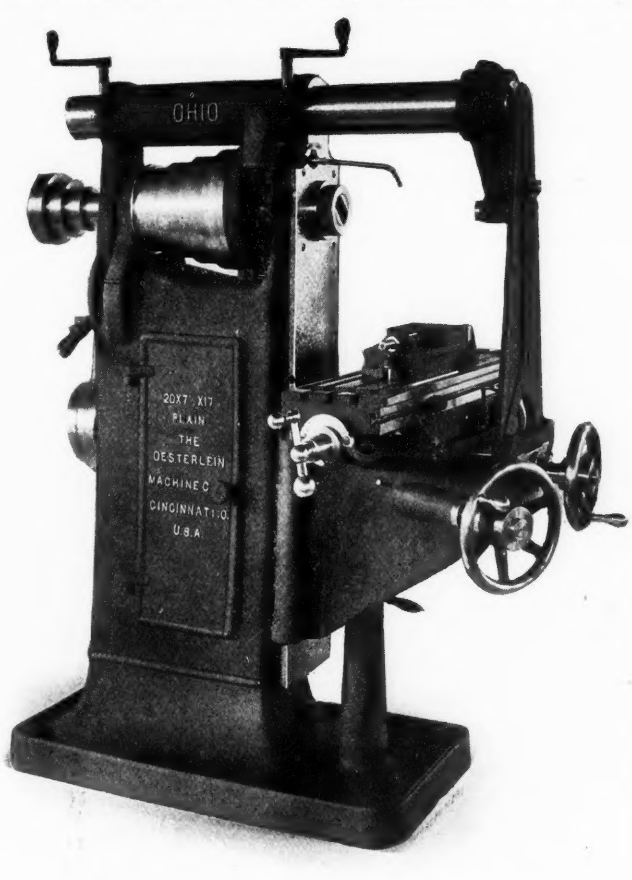


Fig. 2. Rear View of Machine showing Drive to Spindle and Feed

viding an ample distance between centers. The saddle has power quick return and by the use of a reversing motor or double throw of switch, fast power traverse is available in both directions.

THE OESTERLEIN NO. 20 PLAIN MILLING MACHINE

The Oesterlein Machine Co., Cincinnati, Ohio, has just placed on the market a No. 20 plain milling machine which is identi-



Oesterlein No. 20 Plain Milling Machine

cal with the No. 20 universal machine built by this company, except in the design of the saddle.

These machines have an improved type of cone which provides a larger belt contact than that customarily found on machines of this size. The back-gear is placed inside the column of the machine below the spindle, so that it is completely enclosed. The column is provided with oil wells from which the spindle is lubricated. The knee is locked with a taper sliding gib; it is clamped along the entire face of the column, and operated by means of a single lever. The arbor is driven by means of a clutch at the front of the spindle, and the design of the spindle nose duplicates that on the larger sized machines built by this company, so that all tools and cutters are interchangeable.

MOTOR DRIVE FOR CINCINNATI GRINDER

The accompanying illustrations show a new style of motor drive which has been developed by the Cincinnati Grinder Co., Cincinnati, Ohio, for application on all sizes of its standard

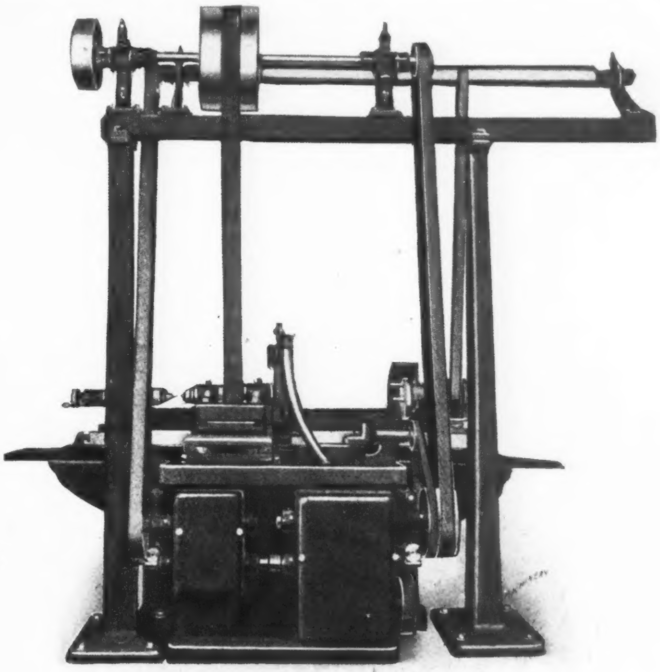


Fig. 1. Cincinnati Grinder showing Drives to and from Gear-boxes

universal machines. Referring to the illustrations it will be seen that the motor is placed on the floor at the right-hand end of the machine and that the overhead works are supported by pedestals bolted to the floor. This arrangement enables a

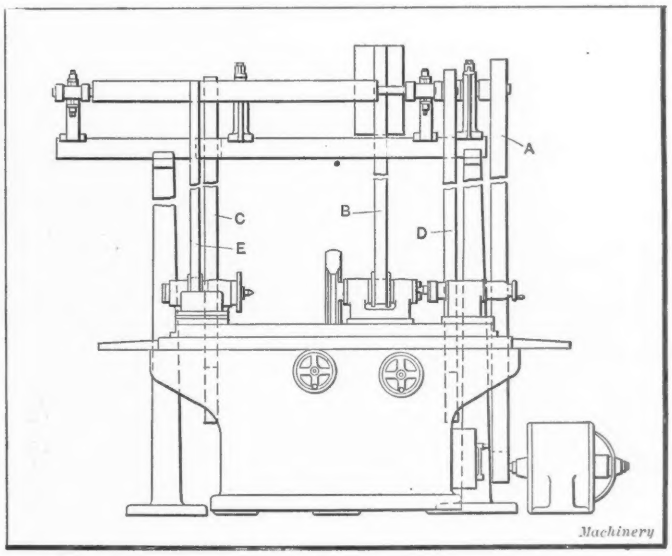


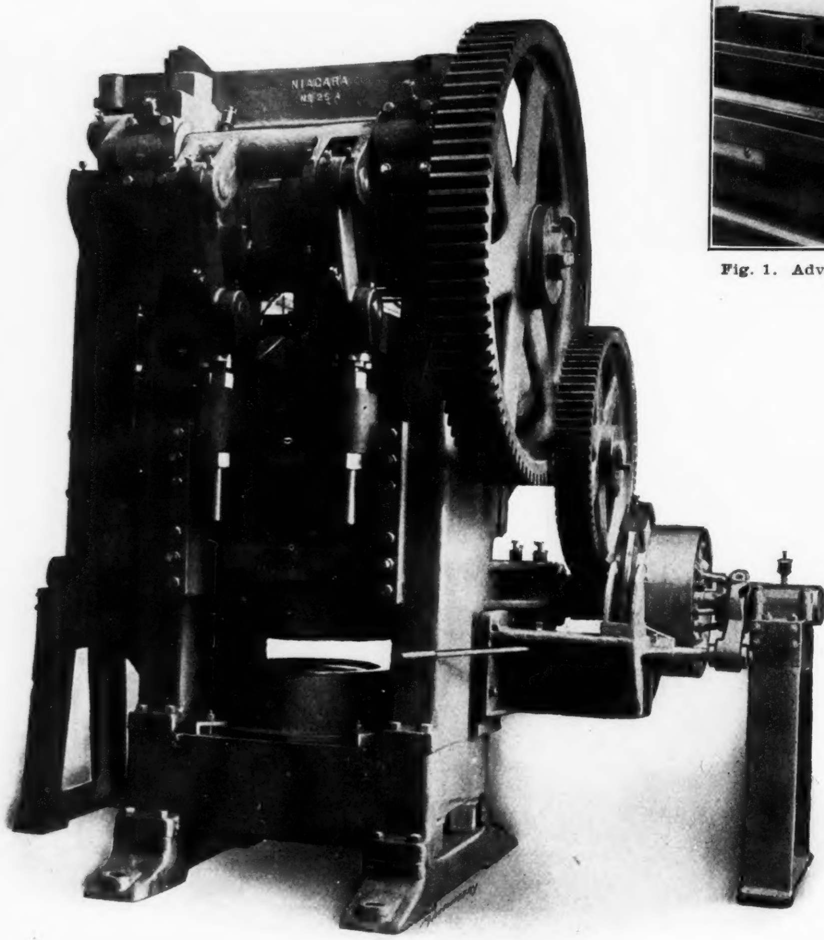
Fig. 2. Arrangement of Motor Drive for Cincinnati Grinder

crane to be used for lifting work on or off the machine, and also provides the required clearance to enable a crane to pass over the machine.

Referring to the illustration Fig. 2, the single-speed double belt *A* transmits power from the motor to the driving pulley on the overhead works. The single belt *B* runs over a drum

on the driving shaft and transmits power to the wheel spindle. The single belt *C* connects with the feed-box at the rear of the machine, as shown in Fig. 1, and the changes of feed obtained from this box are transmitted by a second belt running through to the front of the machine.

Direct connection is made between the feed- and speed-box as shown in Fig. 1, and the changes of speed at which the work is rotated are transmitted to the driving drum on the overhead works by means of the belt *D;* power is then taken down to the work spindle by the belt *E*. An advantage of the speed changes secured by the gearing on the machine lies in



Niagara Double-action Drawing Press

the fact that the travel of the table is entirely independent of the work speed, or speed of the wheel, so that the work may be traversed the full width of the wheel face at each revolution. This feature is of particular value when the machine is required to remove metal as rapidly as possible.

NIAGARA DRAWING PRESS

The machine illustrated herewith is a double-action toggle drawing press which was recently built by the Niagara Machine & Tool Works, Buffalo, N. Y. This press is of particularly massive construction and is adapted for a large variety of heavy work of the kind met with in factories engaged in the manufacture of automobile parts, enameled iron ware, and other articles of a similar nature.

The frame of the machine is of the built-up type held together by four steel tie rods. The blank holder is operated by four powerful toggles which receive their rocking motion from a pitman and slide on the left-hand side of the machine. The two horizontal rocker shafts are made of steel and are timed to give the blank holder the required "dwell" during the drawing period. The plunger or inner slide receives its motion from a pitman from the main shaft. The press is equipped with a powerful friction clutch actuated by a hand lever

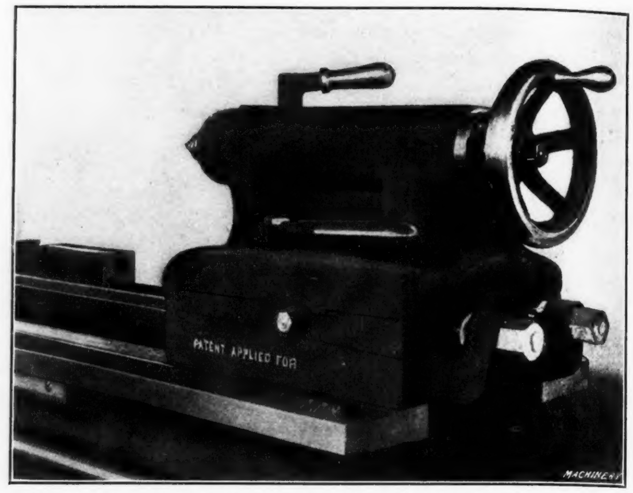


Fig. 1. Advance Tailstock with means of adjusting for Wear in Vees

which provides for stopping and starting the machine at any time during the stroke. This clutch is of the multiple disk type and has proved to be particularly suitable for use on presses of this class. The outboard bearings for the pulley shaft are provided with a double wedge adjustment to obtain perfect alignment when mounting the press on its foundation. The press has a capacity for blanks up to 30 inches in diameter and a maximum depth of 8½ inches. The stroke of the plunger is 18 inches and the weight of the machine about 41,000 pounds.

ADVANCE ADJUSTABLE TAILSTOCK

In the patented tailstock made by the Advance Machine Tool Co., 591 Twenty-second St., Milwaukee, Wis., and shown in the accompanying illustrations, means have been provided for making vertical adjustment to compensate for wear in the tailstock vees. Referring to the illustrations, it will be seen that a taper wedge is placed beneath the tailstock, and that there are two studs inserted at the back of this wedge. Two lugs are cast to the end of the base, through which the studs move freely, and two nuts are provided on each stud, one nut being at each side of the lugs through which the studs pass. These

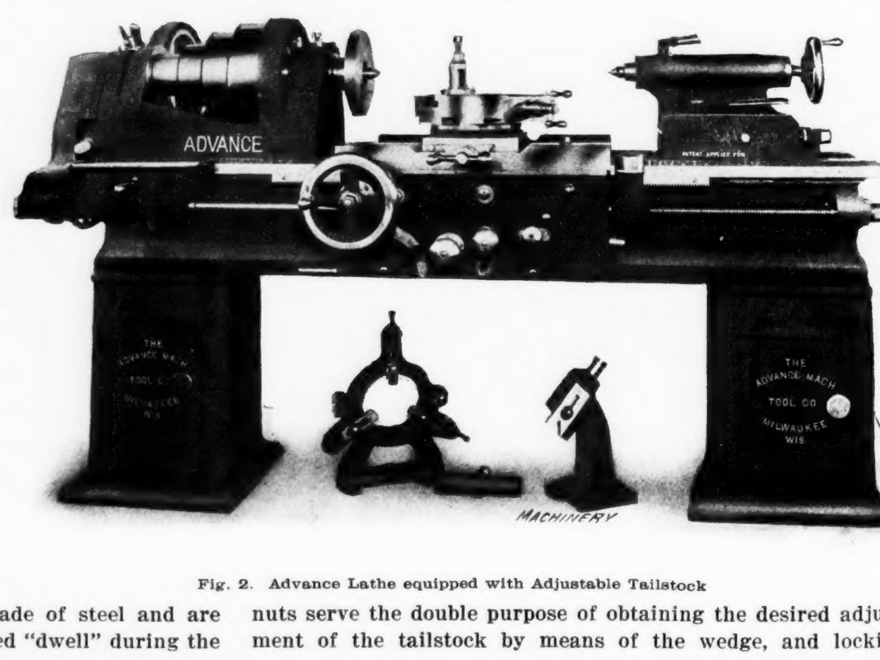


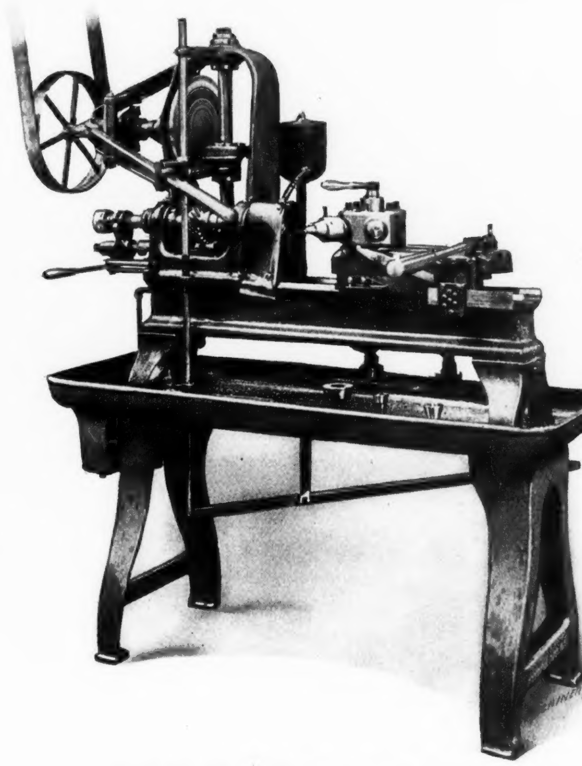
Fig. 2. Advance Lathe equipped with Adjustable Tailstock

nuts serve the double purpose of obtaining the desired adjustment of the tailstock by means of the wedge, and locking

the wedge in place when this adjustment has been obtained. A tongue and groove maintain alignment between the base and the wedge, and the rigidity of the tailstock is not impaired in any way through the provision of this means of adjustment. A vertical adjustment of ⅛ inch is available by this method, the taper wedge and base having a taper of ½ inch to the foot. A new style of single back-geared lathe of this company's manufacture is shown in Fig. 2 equipped with one of these tailstocks. The same type of lathe is also made with double back-gears.

WELLS FRICTION HEAD HAND SCREW MACHINE

A hand screw machine equipped with friction drive, which has recently been placed on the market by F. E. Wells & Son Co., Greenfield, Mass., is illustrated herewith. The arrangement of the friction drive on this machine will be readily understood by reference to the accompanying illustration, where it will be seen that the power from the single pulley is transmitted through the friction disks to a spiral gear on



Wells Hand Screw Machine with Friction Head

the spindle of the machine. The advantages which the manufacturers claim for this machine may be briefly outlined as follows: Quick changes may be made from one speed to another, and quick reverse for running off when the machine is engaged in tapping or threading operations. The friction drive makes it possible to get a greater variety of speeds and to make the changes more quickly than can be done by any other method. The machine is driven by either a plain countershaft or direct from an electric motor.

NEW MACHINERY AND TOOLS NOTES

Flexible Coupling: Francke Co., New Brunswick, N. J. A flexible coupling which is similar in appearance to the regular flange type but with the flanges connected by flexible pins instead of rigid bolts. This arrangement enables the coupling to adjust itself for slight errors in shafting alignment.

Knee Drilling and Tapping Machines: Sloan & Chase Mfg. Co., Newark, N. J. Drilling and tapping machines in which the height of the table is regulated by the knees, leaving both of the operator's hands free to operate the machine and handle the work.

Set-Over Turret: Acme Machine Tool Co., Cincinnati, Ohio. A set-over turret designed for outside turning, facing, boring and recessing with a single point tool. The set-over movement is operated by a ball crank and screw which is provided with a micrometer dial.

Portable Steel Hoist: Lafayette Engineering Co., Lafayette, Ind. A portable hoist constructed of standard structural steel sections, which has a height of 8½ feet and an overhang of 3½ feet from the upright to the hook. This hoist is especially adapted for handling all kinds of work met with in a machine shop.

Shear Blade Grinder: Springfield Mfg. Co., Bridgeport, Conn. A shear blade grinder which has a capacity for work up to 130 inches long by 8 inches wide and 3 inches thick. The knife holder is supported by bearing brackets at intervals of 30 inches and is locked at each end. Positive locking stops are provided to facilitate resetting the holder.

Engine Lathe: Reed Hardware & Mfg. Co., Cairo, Ill. This company has improved its engine lathes, making the design heavier throughout. A set-over screw and feed pinion have been applied to the tailstock, and the tool block and compound rest have been improved to make them better suited for heavy classes of work.

Belt Driven Grinding and Polishing Machine: Excelsior Machine & Tool Co., East St. Louis, Ill. This company has recently added to its line three types of swing frame grinding and polishing machines and a surface grinding and polishing machine. Two of the swing frame machines are designed for motor drive, while the third is a belt-driven machine.

Vertical Air Compressor: Ingersoll Rand Co., New York City. A line of small vertical compressors provided with an automatic lubricating device, simple plate and ring valves, and enclosed dust-proof construction. The machines are built in several different sizes, one of which is adapted for portable service when mounted on a truck designed for this purpose.

Portable Air Compressor: Brunner Mfg. Co., Utica, N. Y. A portable air compressor designed for cleaning machinery and for other general classes of shop use. This compressor is driven by an electric motor which can be operated from any lamp socket. It has a sufficient capacity to enable it to operate for from 10 to 20 minutes from the storage supply.

Portable Shear: Danville Foundry & Machine Co., Danville, Pa. This company has recently modified the design of the "Little Giant" portable shear. It is operated by one horsepower motor and the speed can be regulated from forty to sixty strokes per minute. This shear has a capacity for round stock up to ¾ inch or plates 4 by ⅜ inch.

Motor Driven Lever Feed Disk Grinder: Chas. H. Besly & Co., Chicago, Ill. This company has recently made a number of improvements in its regular line of disk grinders. These include the addition of a geared feed table and electric motor drive, which makes the machine essentially a manufacturing tool adaptable for use in a great variety of metal- and woodworking industries.

Swing Grinder: W. D. Pratt Co., Canton, Ohio. This machine is suspended from the ceiling and the wheel swings on an arm so that it can be readily brought into position. The wheel is located at the side of the operator for safety. In addition to its use as a grinder, the machine can be fitted with a wire brush for cleaning castings and similar classes of work.

Ring Wheel Grinder: Springfield Mfg. Co., Bridgeport, Conn. A ring wheel grinder equipped with a pump and water shield to provide for wet grinding. The machine is driven by a belt on the grinder spindle and the table has a quick rack and pinion hand feed which is operated by a pilot wheel. The cross and vertical movements are operated by smaller handwheels.

Motor Drive for Engine Lathes: Nordberg Mfg. Co., Milwaukee, Wis. This company has recently developed two methods of mounting a motor on an engine lathe for single pulley and cone pulley drive. The motor is supported above the lathe headstock and although the distance between centers is necessarily short, an efficient means of adjusting the belt tension is provided which gives very satisfactory results.

Portable Boring, Drilling and Milling Machine: Newton Machine Tool Works, Inc., Philadelphia, Pa. This machine is driven by an Otis Elevator Co. 7½ horsepower motor running at from 300 to 1200 R. P. M., which gives spindle speeds varying from 8.1 to 150 R. P. M. and feeds of 0.0045 inch, 0.0094 inch and 0.0124 inch per spindle revolution. The machine is particularly adapted for drilling motor frames and other classes of work where it is desired to drill a number of holes at different levels.

Herringbone Gear Planer: Fawcus Machine Co., Pittsburg, Pa. This machine marks a wide departure from preceding equipments for machining herringbone gears. Two tools operate on two halves of each tooth at the same time and the cutting strokes are from the edges of the gear blank to the middle of the face. By this arrangement, the pressure of the tools is neutralized by the gear itself. Any practical helix angle can be cut and the profiles of the teeth are obtained from formers in the same manner as for spur gears.

* * *

When a man begins to be afraid that some other fellow is going to beat him out of his job, he confesses that he is too weak to hold it.

REVERSING MOTOR PLANER DRIVE

APPLICATION OF THE TRIUMPH-MONITOR REVERSING MOTOR DRIVE TO PLANERS

Planers have been driven by individual electric motors for several years, but the motor instead of being directly connected to the planer driving shaft, is generally connected to it by belts. Several electrical motor concerns have been experimenting and developing planer drives which could be directly connected to the planer and be capable of reversing automatically without excessive shock. The Triumph Electric Co., Cincinnati, Ohio, began a series of experiments several years ago with a view of developing special apparatus to meet the requirements of a self-contained motor drive. The controller was first designed to operate the motor used in driving the planer, and afterward special motors were built to suit both the controller and planer.

At present there are what might be called two distinct types of reversing motor planer drives, the difference between them consisting in the method used in reversing the motor. In the first type the motor is started automatically in the usual manner, stopped at the end of the stroke by dynamic braking and then reversed by applying reversed power, this cycle being repeated at the end of each stroke. In the second type, which is that used in the "Triumph" equipment, dynamic braking is dispensed with, and the planer is stopped and reversed by disconnecting the motor and immediately applying reversed power. The advantages claimed for this method are: Increased speed of reversal, greater uniformity in length of stroke and decreased wear on both motor and controller.

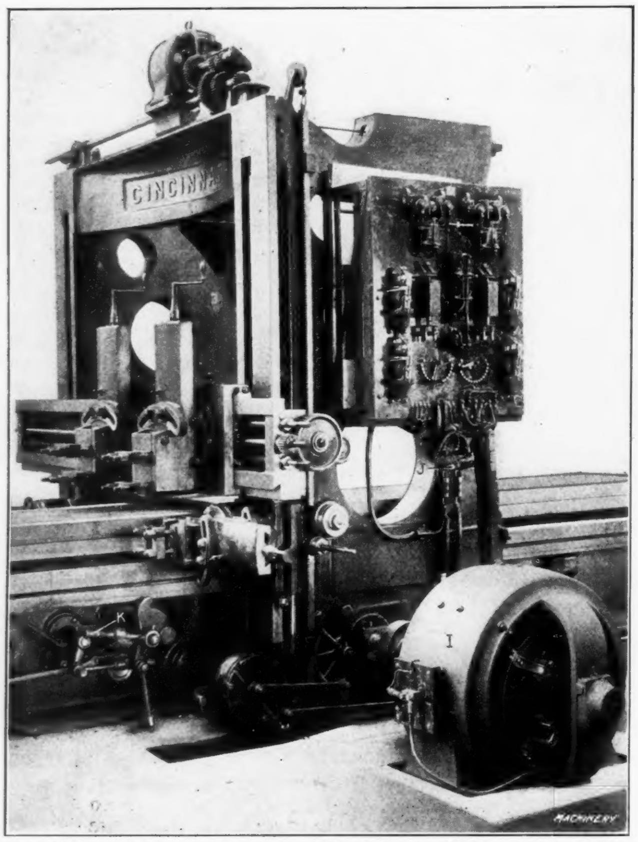


Fig. 1. "Triumph-Monitor" Reversing Motor Drive applied to a 76-inch Planer

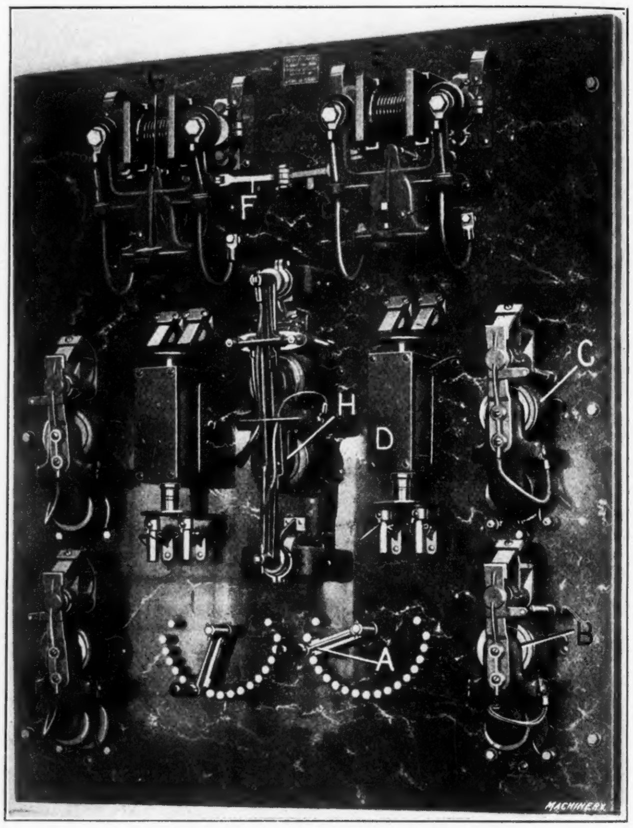


Fig. 2. The "Triumph-Monitor" Controller

Operation of Monitor Controller

One of the chief points of interest in connection with this reversing motor planer drive is the controller which is shown in Fig. 2. This large panel carries the apparatus for automatically accelerating the motor to predetermined speed in the desired direction, and for starting and stopping it at every cycle. It also contains the rheostat handles which control the cutting and return speeds. The right-hand side of the panel carries all the apparatus for operating the planer on the return stroke, while the left-hand side carries the apparatus for controlling the motor for the cutting stroke. Referring to Fig. 2, *A* is the speed regulating rheostat for return stroke, *B* and *C* armature resistance short circuiting switches, *D* accelerating switch for return stroke and *E* switch supplying power to motor for return stroke. The mechanism on the left side performs the same functions for the cutting stroke. The lever *F* prevents the "returning" and "cutting" switches *E* and *G* from operating at the same time. The mechanism *H* in the center of the panel is the main circuit breaking switch.

The complete equipment is shown in Fig. 1 applied to a 76-inch Cincinnati planer. The main driving motor *I*, which in this case is a 30 H. P. compound interpole variable speed motor, drives the planer platen in either direction through the planer gear mechanism. The shaft, connected through gears with the bull wheel, is flexibly coupled direct to the motor armature. This insures that the power generated by the motor is transmitted to the driving mechanism of the planer with the least possible loss of power.

In operating the apparatus, the main line switch *J*, see Fig. 1, is closed; this connects the control panel with the line, but does not start the motor. To start the planer it is necessary to operate the master switch *K*, which replaces the regular tumbler handle used on the belt-driven machines. When this switch is turned toward the "start" position the motor starts. Then as the platen moves in either direction, the stops throw over the tumblers, and these through the table reversing switch *L* operate the switch on the control panel, stopping the rotation of the motor and immediately applying reversed power to rotate the motor in the opposite direction. The planer is stopped by turning the master switch in the "stop" direction. This master switch operates the motor so quickly that it is possible, by turning the switch to the "start" and immediately to the "stop" position, to transmit a movement to the table of ½ inch. The length of the stroke does not vary more than ⅛ inch on cuts of any length or at any speed. This feature is very desirable when planing in pockets and close to ledges.

The sequence of controller operations when starting the motor is illustrated diagrammatically from *A* to *D* in Fig. 3. When the master switch handle is turned to the "start" position, power is thrown on the motor armature with resistance in series. The resistance is then short-circuited and the motor

operates as a compound interpole machine with full field strength. Short-circuiting the series field converts the motor into a shunt interpole machine still retaining full shunt field strength. The shunt resistance short-circuiting switch is then opened, see *D* Fig. 3, which causes the motor to attain its

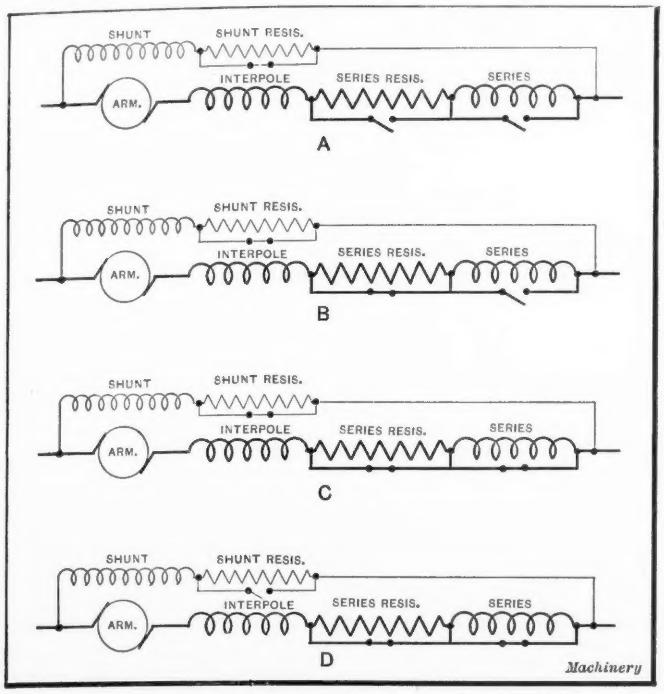


Fig. 3. Sequence of Controller Operations on Starting Motor

predetermined speed. When the dog on the planer platen throws the tumbler, all switches return to their initial positions and power is immediately applied in the reversed direction. As the platen traverses back and forth this sequence of operations is repeated.

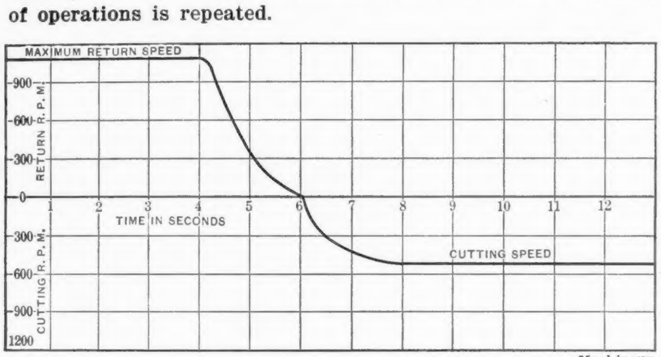


Fig. 4. Curve of Motor Speed on Reversal with Dynamic Brake Equipment

The sequence of steps just outlined follows so rapidly that only one second elapses between the throwing of the tumbler, and the last switch operation for reversed rotation. At any point in the cycle a reversed movement of the master switch trips all the switches on the controller panel and the planer

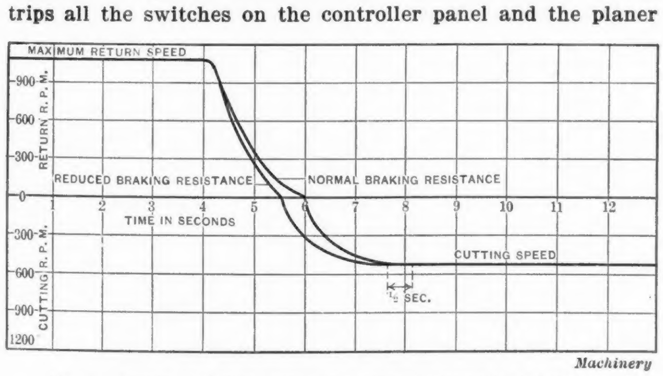


Fig. 5. Curves of Motor Speed on Reversal, Dynamic Brake Type, showing Effect of Reduction of Braking Resistance

is stopped promptly. This service is difficult for the motor, but the motors have been so designed that sparkless commutation for the complete cycle results; in fact the commutator attains that blue-black polish which indicates entire absence of sparking and perfect commutation.

Efficiency of the Dynamic Brake as Compared with the Method of Applying Reversed Power

As previously mentioned two methods are in use for stopping and reversing the motor at the completion of the stroke, and what follows is a comparison of the efficiency of

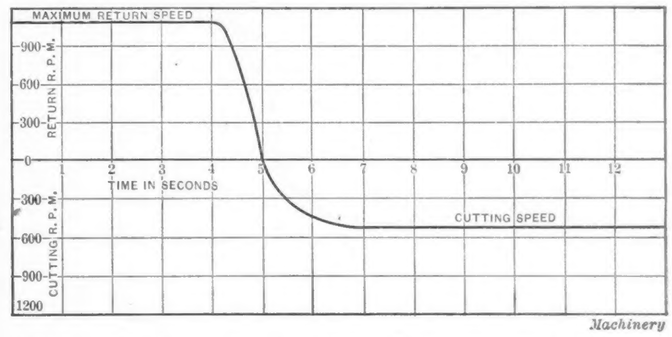


Fig. 6. Curve of Motor Speed on Reversal, "Triumph-Monitor" Equipment

the two methods. The accompanying oscillograph curves illustrated in Figs. 4 to 9 inclusive, show wherein the method of applying reversed power to the motor through resistance

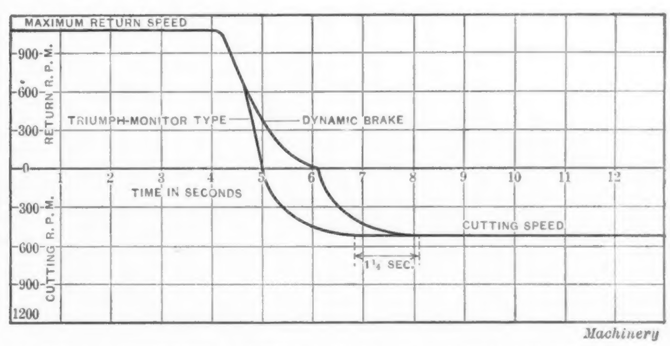


Fig. 7. Comparison of Curves plotted in Figs. 4 and 6 showing Gain in Reversal over Dynamic Braking

for both braking and reversing shows a gain in time over the method of employing a dynamic brake for this purpose. The curve plotted in Fig. 4, is taken with the dynamic brake equipment, and shows the motor speed in R. P. M. when

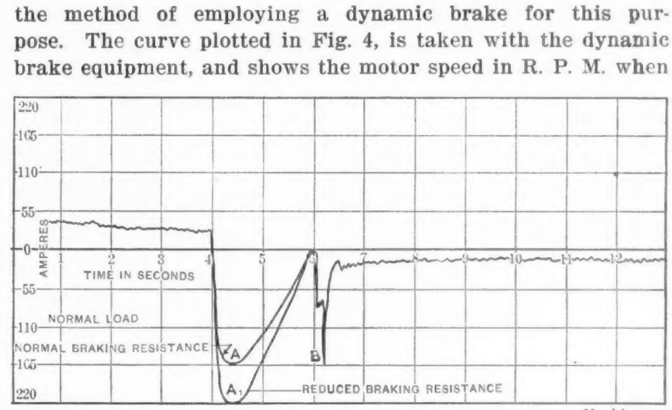


Fig. 8. Curves of Current Consumption on Reversal, showing Effect of Reduction of Braking Resistance

reversing from maximum return to moderate cutting speed. The curves plotted in Fig. 5 show the effect on the speed of reversal by decreasing the dynamic braking resistance which causes a gain in time of reversal of ½ second.

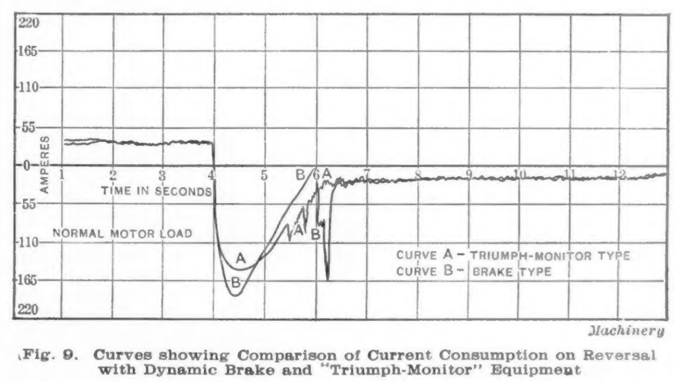


Fig. 9. Curves showing Comparison of Current Consumption on Reversal with Dynamic Brake and "Triumph-Monitor" Equipment

In Fig. 6 is shown the curve obtained from the "Triumph-Monitor" equipment, and in Fig. 7 the curves plotted in Figs. 4 and 6 are arranged together for ease of comparison. As the

time graduations are in seconds, the difference of 1¼ second on the time of reversal is easily observed. In Fig. 8 two curves are plotted to show the corresponding effect on current rush at the setting of the brake, with normal and reduced braking resistance. The peak *A* is raised from 165 to 200 amperes, which latter value is 100 per cent overload for the motor. The second shock *B*, also shown, is due to application of reversed power. The curves illustrated in Fig. 9 show the current required to operate the two types of reversing motors under the conditions indicated in Fig. 7. The two motors which were tested were of the same make, of the same size and rating, and operated under identical conditions.

Record of Tests

The Triumph Electric Co., has made many tests to determine the capacity and capabilities of the reversing type of motor drive and some of these will be given in the following. A test which shows some interesting facts was made on an 84-inch Pond planer with a "Triumph," 40-horsepower, 230-volt motor, operating from 250 to 1000 R. P. M. The controller was 40 H. P. capacity, 230 volts. The material cut was two 0.50 carbon steel billets. Two tools were used with a depth of cut of ¾ inch at ⅛ inch cross traverse feed, and at a cutting speed of 25 feet per minute. The following table gives the data obtained from power readings taken under the conditions specified and also with tools removed.

	Volts	Amperes	Watts	H. P.
Total input on cut	220	150	33,000	44.3
Input, running idle (no cut)	220	16	3520	4.7
Apparent power for cutting	...	...	29,489	39.6
Motor load losses	...	...	1190	1.6
Net power for cutting	...	...	28,290	38.0

The material removed at this rate was 56.3 cubic inches per minute, or 16.3 pounds. The power required to remove a section of one square inch at 25 feet per minute cutting speed would be 203 H. P. The cut is approximately the maximum that should be taken with this equipment, an intermediate cast-iron gear having been broken shortly after the readings were taken.

Another cutting test, which was performed on a 46-inch by 30-foot Gray planer, is charted in Fig. 11. The motor in this case was a "Triumph" 30-horsepower, 115 volts, and operating from 275 to 1100 R. P. M. The controller was a 30-H. P., 115-volt "Monitor." The material cut was wrought iron; depth of cut, ⅝ inch; cross traverse feed, ⅜ inch; and cutting speed, 46 feet per minute. The chart given in Fig. 10 shows the same equipment with the load removed and gives some indication, when compared with the curve in Fig. 11, of the actual power required to take the cut with two tools in operation on separate bars at the same time.

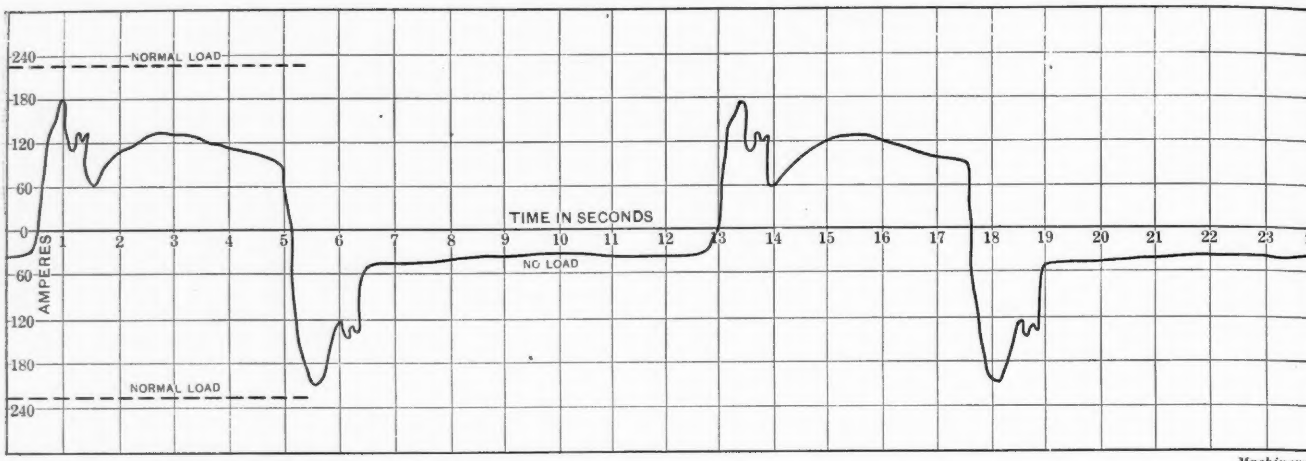


Fig. 10. Curve of Test made on a 30 H. P. "Triumph" Reversing Motor applied to a 46-inch by 30-foot Gray Planer running Light

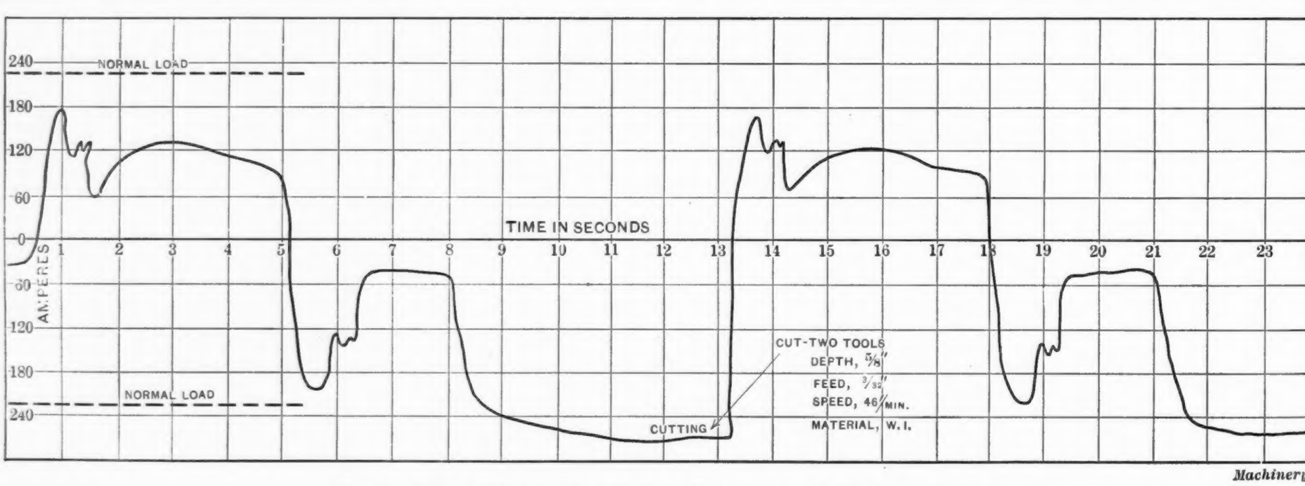


Fig. 11. Curve of Test made with Same Equipment as in Fig. 10 under Cut

Reversing Motor Drive Compared with Ordinary Belt Drives

There are several points of superiority claimed for the reversing motor drive over the ordinary belt drive, among which may be mentioned the following: Quantity and quality of finished product, flexibility, appearance, quietness, and cleanliness.

There is no doubt that this type of drive is capable of transmitting more power than the usual belt type, and will remove more metal in a given time, provided the planer is designed to stand the maximum cut. It is stated that the work shows a smoother, cleaner cut because of reduced vibration on the planer housings. The vibration is minimized by placing the motor on the floor instead of five or eight feet above the work. It is also claimed that the reversing motor drive gives a much more uniform length of stroke, and enables planing to a line ⅛ inch wide completely across the platen. Edging may also be accomplished in ½ inch strokes.

As to efficiency, it is claimed that on planers above 30 or 35 inches width the reversing motor drive shows a perceptible gain, while for planers below these sizes the efficiencies of the belt and reversing motor type of drive are about equal. Where the motor drive shows the greatest efficiency is on heavy work, and it is capable of taking tremendous cuts, the strength of the planer being the limiting factor. This type of drive is also being applied to vertical slotting machines with good results.

BRISTOL RECORDING DIFFERENTIAL PRESSURE GAGE

A line of recording differential pressure gages has been developed by the Bristol Co., Waterbury, Conn. Some of these recorders have been in service continuously since the preliminary models were first sent out in 1908, and the design and construction of the instruments that are now being placed on the market is based on results obtained in actual service during the last four years. These recording differential pressure gages are designed for use in connection with venturi meters, pitot tubes, orifices, combinations of orifices, etc., to record velocities and volumes of air, gas, steam, water and other liquids flowing through mains and pipes. These recorders may also be used to advantage for recording differences and variations of liquid level in steam boilers, pressure tanks, filter beds, process kettles, etc.

Fig. 1. Bristol Recording Differential Pressure Gage

Patents have been granted to W. H. Bristol, president of the Bristol Co., covering novel features resulting from his work in developing differential recorders. The fundamental principle employed in the construction of these differential pressure gages is that one pressure is applied to the inside of the operating tube, while the other is applied to the outside of the same pressure tube within a closed casing. In order to record the movement of the pressure tube, it becomes necessary to transmit its motion to the outside of the pressure tube casing. As the differential pressure to be recorded is usually small, as compared with the static pressure, the operative force is correspondingly small, and it is quite evident that it will be impractical to use a stuffing-box around a shaft passing through the pressure casing on account of the friction which would be produced. To avoid the use of a stuffing-box, a unique frictionless sealing device is employed, as described in detail below.

Fig. 1 shows the exterior of one type of these recorders. Figs. 2 and 3 show part of the interior construction of the spring tube type of differential recorder. In Fig. 2, the pressure tube *A* is of the helical type which has been successfully used for many years in Bristol recording pressure and vacuum gages. One of the pressures, the difference of which is to be recorded, is applied to the interior of this tube through the pipe *B* and the other pressure is applied to the exterior of the tube through the pipe *C*. It will be noted that the pressure tube *A* is entirely enclosed in the pressure tight casing *D*, and that the movement of the pressure tube will be in proportion to the difference between these two pressures. This motion is transmitted to the recording pen arm *E* (outside of the pressure casing), by means of a small shaft *F* passing through the long tubular sleeve *G*. The capillary action of the oil or liquid between the sleeve and shaft makes this joint both frictionless and pressure tight.

In Fig. 3, *A* is the pressure tight casing enclosing the diaphragm pressure tube *B*, and in a similar way one pressure communicates with the interior and the other pressure with the exterior of this tube, and its motion is transmitted by means of the rotating shaft through the sleeve *C* to the recording pen arm *D*. The length of this sleeve is many times the diameter of the shaft passing through it, differentiating it from an ordinary bearing. This ingenious device permits re-

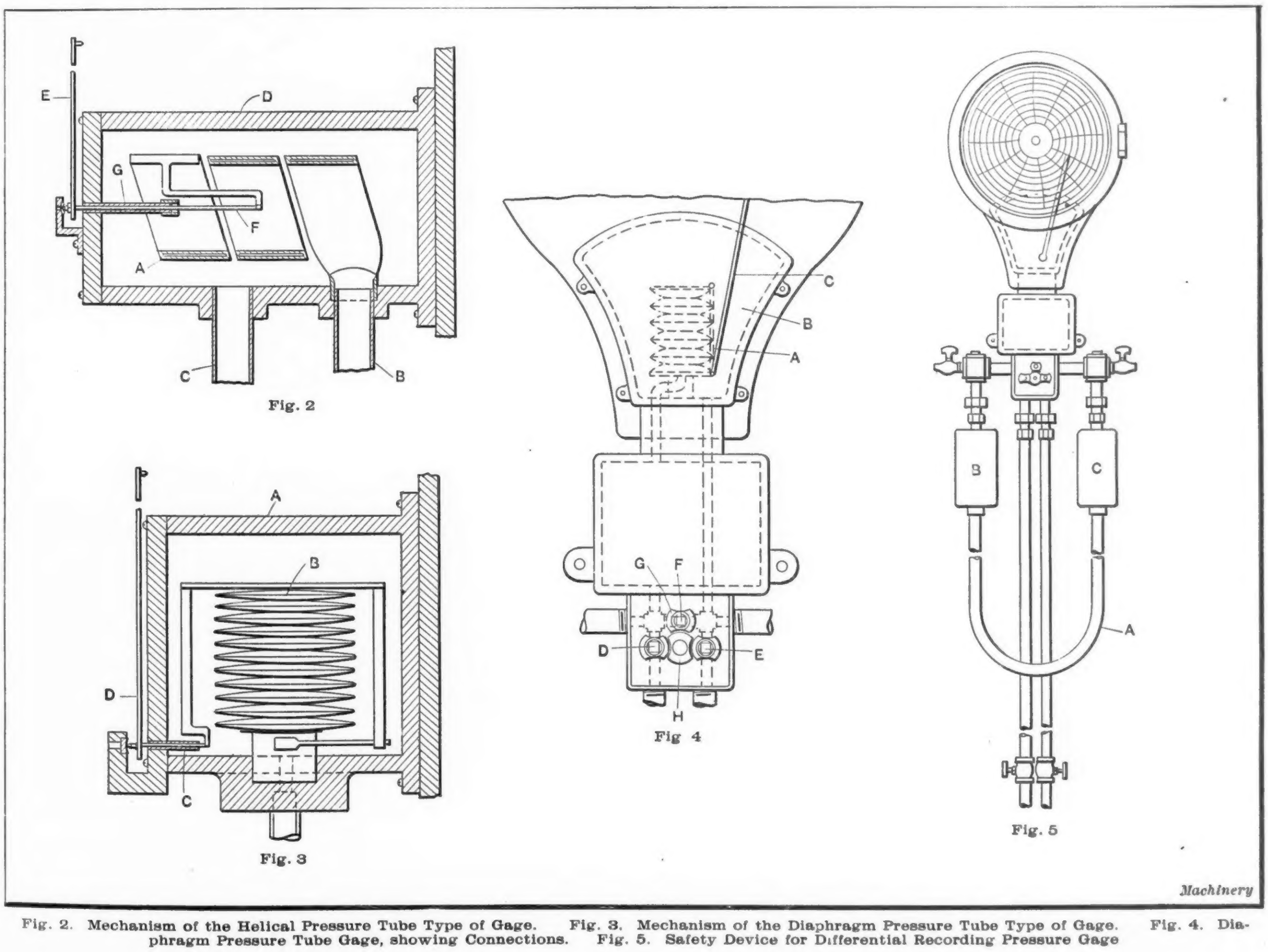


Fig. 2. Mechanism of the Helical Pressure Tube Type of Gage. Fig. 3. Mechanism of the Diaphragm Pressure Tube Type of Gage. Fig. 4. Diaphragm Pressure Tube Gage, showing Connections. Fig. 5. Safety Device for Differential Recording Pressure Gage

cording extremely small differences between the respective pressures existing inside and outside of the pressure tube. It has been found that the simple frictionless sealing sleeve through which the pen arm shaft passes does not produce appreciable resistance to the rotation of the shaft, and at the same time capillary attraction and adhesion prevent leakage of even high pressures from the pressure casing.

In Fig. 4, the diaphragm tube *A* is shown in the pressure chamber of the casing *B*, directly connected to one end of the rotating shaft which passes through the pressure sleeve, and has its other end directly connected to the recording pen arm *C*. This diagram also shows a set of interlocking valves *D*, *E* and *F*, which constitute a device for adapting recording differential pressure gages to practical operating conditions. The cross valve *G* is shown in an open position connecting the two pressure pipes. Valves *D* and *E* in the pressure pipes are shown closed. Both of these valves *D* and *E* can be opened, allowing the static pressure from either pipe to be applied simultaneously to the inside and the outside of the pressure tube of the actuating mechanism. The interlocking member *H* can then be turned through an angle of 90 degrees, thus making it possible to close valve *G* which completes the connections, so that the difference in pressures is recorded.

Fig. 5 shows a patented safety device consisting of a "U" shaped tube *A* partly filled with a suitable liquid, such as mercury or water. This tube has enlargements *B* and *C*, each having sufficient volume to accommodate the quantity of liquid contained in the "U" tube. The length of the tube varies with the range of the gage so that the greatest possible head of the liquid contained in the tube corresponds with the total range of differential pressure that the gage is designed to record. Should the full static pressure be admitted by accident to either side of the differential gage, the liquid contained in the safety "U" tube would instantly be forced up into one of the enlargements, thus allowing the static pressure to be applied simultaneously to both inside and outside of the pressure tube and protecting it from being destroyed.

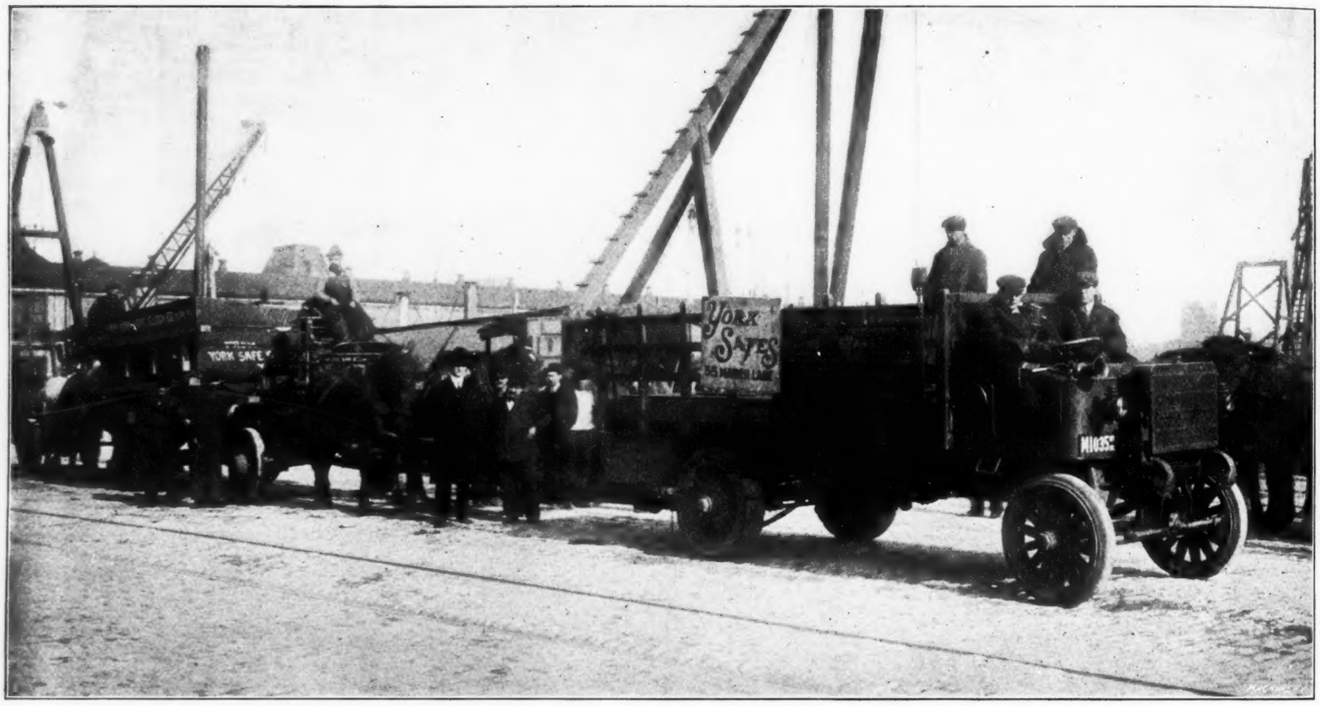
* * *

The Isthmus of Panama was first crossed by Balboa in 1513, and as early as 1550 the question of digging a canal was first raised by the Portuguese Galvoa, who published a book to demonstrate that a canal could be dug at Nicaragua, Darien or Panama. The Spanish Government, however, which had been urged to undertake the work, was opposed to it. In 1843, the French Government advocated a canal, and the first complete surveys of both the Nicaragua and Panama routes were made in 1872 to 1875 by the United States Government. The French began the actual work on the canal in 1881 and discontinued in 1889 for lack of funds.

ANOTHER HEAVY LA FRANCE HYDRAULIC TRANSMISSION TRUCK HAUL

An illustration and brief description was published in the February number of MACHINERY of a 45-ton load, hauled by a La France hydraulic transmission truck (sold by the Hydraulic Truck Sales Co., New York City). The accompanying illustration shows another heavy load hauled March 18 in New York City, the load being part of heavy vaults made by the York Safe & Lock Co. for the bank clearing house. The frame for the door of the new vaults had to be moved through the streets. The frame was brought to New York on a lighter and delivered to Pier 11, East River. It was then loaded on a four-wheel wagon which itself, empty, weighs 16,400 pounds. The frame, which is 7 feet 6 inches wide, 9 feet long and 3 feet 6 inches thick, weighs 52,600 pounds. The La France hydraulic truck was loaded with five steel plates for the vault, which in the aggregate, weigh 12,100 pounds. The truck itself weighs 4½ tons, the total load to be moved thus being 90,100 pounds.

The shortest route to deliver the load to its destination would have been through Wall St., but permission to use Wall St. was denied because the street paving would not support the load. The truck had to proceed down Front St. to South Ferry and then up through Whitehall St. to Bowling Green and up Broadway to Cedar St. Two horses were hitched to the pole of the truck so as to control the pole and keep the trailer steered right.



La France Hydraulic Transmission Motor Truck hauling Bank Vault Parts. Total Weight hauled, 90,100 pounds

In order to show the remarkable tractive effort that can be produced by the hydraulic transmission gear, the truck was stopped with load on a 4½ per cent grade at Bowling Green and was started again with no difficulty. The draw-bar pull required to start the load and keep it moving up this hill was a little over 9000 pounds.

The speed at which the load was hauled was four miles an hour. It was estimated that at least twenty horses would have been required and that probably it would not have moved over 2½ miles an hour at the most. The probability is pointed out that the load moving at this rate damaged the street less than if it had been moving slower. The load was so heavy that after it had been delivered to the destination, the weight caused the wheels to cut deeply into the asphalt although when moving they had made little or no impression on it. The only damage done to the streets on the route was breaking a small manhole cover.

* * *

The man who tries to keep all his own trade knowledge to himself should remember that he has no right to the ideas of others, if he is too stingy with his own.

RESUSCITATION FROM ELECTRIC SHOCK

The following is a method of resuscitating victims of electric shock which has been developed by Dr. C. A. Lauffler, medical director of the Westinghouse Electric & Mfg. Co. The method may be sub-divided into four operations and one of its important features is that any man with common intelligence can apply it successfully. In addition to its application in cases of injury from electric shock, this method can also be successfully used in resuscitating victims of apparent drowning, persons who have been rendered unconscious from inhaling noxious gases, and similar accidents. The steps in the use of this method are as follows:

The Position of the Face.—The man is laid upon his stomach with his face turned to one side so that the mouth and nose do not touch the ground; his arms are then extended above his head. This causes the tongue to fall forward so that the entrance to the wind pipe is not blocked. It will be evident from the preceding that it is possible for one man to use this method in an attempt to save the life of a comrade if there is no other assistance at hand. In this connection it should be mentioned that it is of particular importance to start to revive a patient as soon as possible after the occurrence of the accident.

Position of the Operator.—The man who is working upon the patient either kneels straddling the patient's thighs or else kneels at either side of the thighs facing the patient's head. The patient's coat and shirt are then removed and the operator feels with both hands to locate the position of floating ribs. He then places one hand on either side of the patient's body so that the little fingers curl over the ends of the twelfth rib and the heel of the hand rests near the ends of the lowest ribs; the hands are placed at some distance from the spine so that a considerable leverage is obtained on the floating ribs.

Mode of Operation.—The operator's arms are held straight so that he applies his full weight from the shoulders by bringing his body and shoulders forward. This force is gradually increased until at the end of three seconds, the pressure upon the lower ribs of the patient is felt to be heavy enough to compress the lungs; the pressure is then suddenly removed, to allow the lungs to expand and draw in a fresh supply of air.

Rate of Respiration Per Minute.—The natural rate of breathing is twelve to fifteen times per minute, and the rate at which this artificial operation of respiration is conducted should not exceed this figure. It is important for the lungs to be thoroughly emptied and about three seconds of pressure is required to do this; the lungs will then be refilled in approximately two seconds. The pressure and release of pressure thus occupies a period of five seconds.

Many victims of electric shock and similar accidents have been saved by this method. The condition of such persons is apparently that of death at the time that the work of resuscitation is started, but where life can be saved there will usually be signs of returning animation in about twenty-five minutes. In any case, however, the operation should be continued for at least an hour and a half and indefinitely if there is any sign of recovery.

* * *

ENGLISH AS SHE IS SOMETIMES WRITTEN

A prominent machine tool builder recently received the following letter from a prospective customer:

"Dear Sir:—I'm glat to see your booklet—to day— Prmete me entroudouc to you my prpese. if you parmete me I tel you something. I net faar steam mill michine for my cantoroy—I want to now. Have you any steam mill michine and yous stone the hours cary of the michine you now why? My cantry never gos Fran and agan I forget I want the steam mill michine 5. hours phors. Plase answer me as soon as posebbel you can. Your varey troulay——"

The following is a "translation" of the letter above:

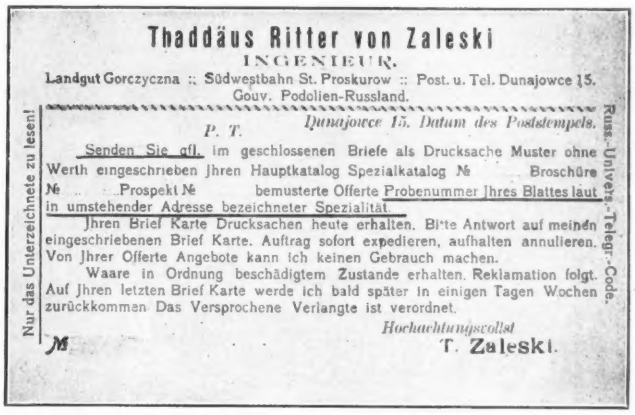
"Dear Sir:—I am glad to have received your booklet, to-day. Permit me to introduce to you my purpose. If you permit me I will tell you something. I need four flour mill machines, steam power driven, for my country. I want to know if you have any steam flour mill machines that use stones. Our people are afraid of machinery. Do you know why? Because my country never goes forward. Now, again, I forgot to say that I want the steam flour mill machines to be of five horsepower each. Please answer as soon as possible.

Yours very truly,"

LABOR-SAVING CORRESPONDENCE SCHEME!

The postcard shown in the accompanying illustration, lately received from Russia, is reproduced with the translation as an example of "efficiency" gone astray. Our German friend has ingeniously solved the problem of formulating requests, acknowledgments, etc., without writing; he simply underscores the words and sentences required to indicate his wishes.

We don't know how effective this ingenious system of correspondence is, but suggest that if correspondence is to be



Thaddäus Ritter von Zaleski

INGENIEUR.

Landgut Gorczyczna :: Südwestbahn St. Proskurow :: Post. u. Tel. Dunajowce 15.
Gouv. Podolien-Russland.

Nur das Unterzeichnete zu lesen!

Russ.-Univers.-Telegr.-Code.

Dunajowce 15. Datum des Poststempels.

P. T.

Senden Sie gfl. im geschlossenen Briefe als Drucksache Muster ohne Werth eingeschrieben Jhren Hauptkatalog Spezialkatalog № ... Broschüre № ... Prospekt № ... bemusterte Offerte Probenummer Jhres Blattes laut in umstehender Adresse bezeichneter Spezialität.

Jhren Brief Karte Drucksachen heute erhalten. Bitte Antwort auf meinen eingeschriebenen Brief Karte. Auftrag sofort expedieren, aufhalten annulieren. Von Jhrer Offerte Angebote kann ich keinen Gebrauch machen.

Waare in Ordnung beschädigtem Zustande erhalten. Reklamation folgt. Auf Jhren letzten Brief Karte werde ich bald später in einigen Tagen Wochen zurückkommen Das Versprochene Verlangte ist verordnet.

Hochachtungsvollst

T. Zaleski.

conducted with the sole idea of saving labor, it be abandoned altogether. Then he will save it all.

Translation of post card from Thaddäus Ritter von Zaleski:

Read Only Portion Underlined

Dear sir:

Please send in sealed letter as printed matter marked sample without value your main catalogue special catalogue No. ... circular No. ... prospectus No. ... offer sample copy of your journal to the above address.

Your letter post card printed matter received today. Please reply to my letter post card. Immediately fill hold up cancel order. I can make no use of your offer proposition. Shipment received good damaged condition. I will send come back to letter in a few days weeks reply to your last letter post card. Promise has been kept request complied with.

Yours truly,

T. Zaleski

* * *

COOPERATIVE INSURANCE

One help in the solving of industrial problems is the creation of more friendly relations between employer and employe. The Foreman's Association of the Stetson Shoe Co., South Weymouth, Mass., has recently arranged to provide a five-hundred-dollar twenty-year endowment insurance policy for each of its twenty-five members through the insurance departments of the savings banks. The company has offered to pay one-half the cost of the premiums on these policies so long as the policy-holder remains in the employ of the company. Each year of service adds to the cash surrender value of the policy and at the end of twenty years, $500 will be paid each member of the association. This act not only protects the policy-holder and his family, but also creates a new incentive for continuing service which helps to hold valuable employes loyal to the company. In time, perhaps, this offer of cooperative life insurance will be extended to other employes. This plan probably will be adopted by others, who, in the interest of loyal service, wish to develop a business-like cooperation between employer and employe. Massachusetts Savings Bank Life Insurance, which through the cooperation of the state and the savings banks furnishes the best insurance at lowest cost, provides an unexcelled opportunity for business men to develop this cooperative insurance with their employes.

* * *

From being one of the most unhealthful and disagreeable jobs in factories the polishing of metals has been changed by the exhaust system to one of cleanliness and fairly agreeable working conditions. All grinding dust and debris is drawn away as fast as produced and discharged into the outer air, making the grinding room one of the best ventilated rooms.

LONG LINES OF MACHINE TOOLS ON

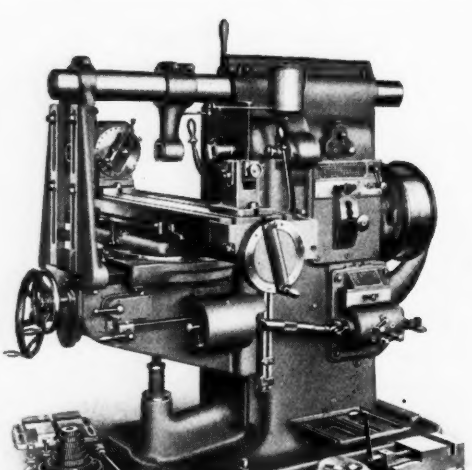
Pictures below give but a partial impression of the space devoted
1,020,900 square feet or about 23½ acres of floor space

PART OF ONE MILLING MACHINE ERECTING FLOOR

The No. 3-A Heavy Universal Milling Machine

a representative example of our extensive line of universal, plain and vertical spindle milling machines **for Heavy Service,** is shown below.

This machine is of the heavy Constant Speed Drive type that was originated at our works, and is well suited to heavy duty in Locomotive Shop Tool Rooms, or in other places where the services of a heavy universal machine are required. It has abundant power for all purposes. It is very rigid. Note the massive appearance of all parts and the stiff support for cutter arbor. Spindle speeds and table feeds are independent. Speed and feed changes are quickly and easily made. The machine can be belted directly to main line. All controlling levers and handwheels arranged for convenience of operator.



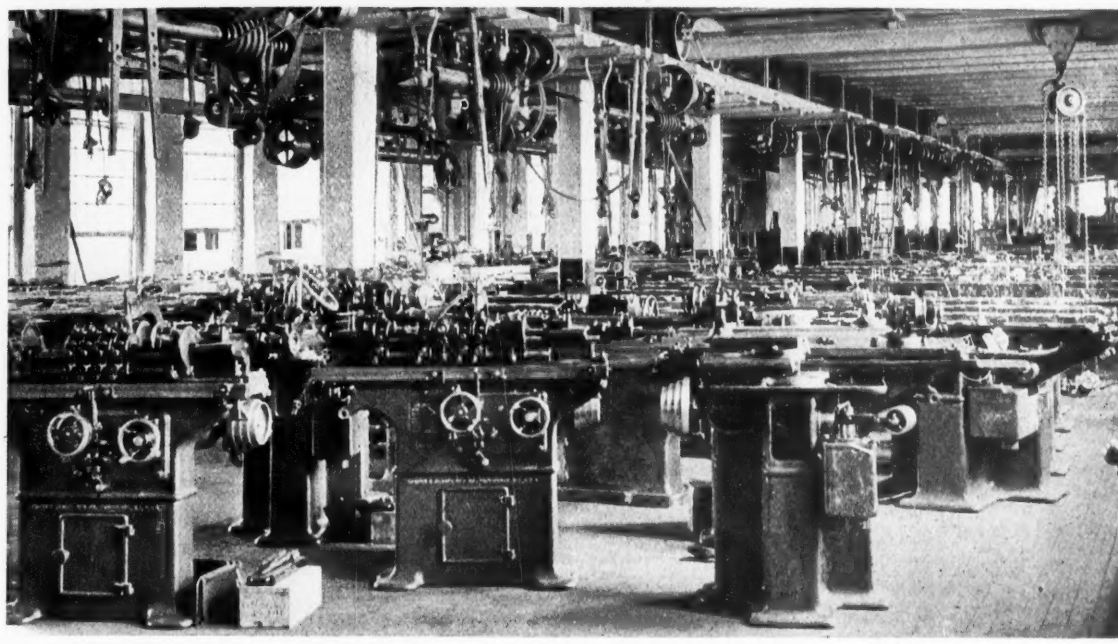
Capacity—Longitudinal feed, 30″; transverse feed, 10″; vertical feed 19″.

Write for special circular of this machine.

BROWN & SHARPE

ERECTING FLOORS IN B. & S. SHOPS

to the manufacture of machine tools. The B. & S. works contain on a large part of which precision machine tools are built.



VIEW ON A GRINDING MACHINE ERECTING FLOOR

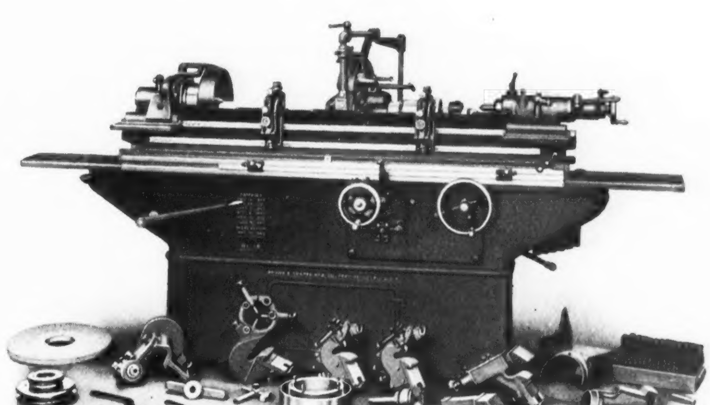
The No. 14 Plain Grinding Machine

shown below is one of our larger sizes, built for producing accurate work when operated under heavy cuts at fast speeds and coarse feeds. It is very rigid, and all alignments are accurately made. The wheel speeds and work feeds are independent. Universal Back Rests and the Automatic Cross Feed for wheel slide insure accurate work and enable more than one machine to be operated by one man on most work.

The machine takes 48″ in length and swings 10″ in diameter.

Do these points interest you?

Special circular on request.



We also manufacture Universal, Plain, Surface, Internal, Cutter and Tool Grinding Machines.

MFG. CO. Providence, R.I., U.S.A.

See page 81

PERSONALS

Edgar N: Easton has joined the railroad sales department of Joseph T. Ryerson & Son, Chicago, and will be located at New Haven, Conn.

Frank Burgess of the Boston Gear Works, Norfolk Downs, Mass., started March 8 for Jacksonville, Fla., to spend a four-weeks' vacation.

C. F. White, formerly tool designer for the Hersey Mfg. Co., South Boston, Mass., manufacturers of water meters, is now employed by the Standard Meter Co., of Brooklyn, N. Y.

F. O. Wells, president of the Wells Bros. Co., Greenfield, Mass., sailed for Europe March 8 for a two months' business and pleasure trip. The European business of the company is growing rapidly.

John H. Craigie, formerly associated with the mechanical department of the Boston & Maine R. R., has joined the railroad sales department of Joseph T. Ryerson & Son, Chicago, his headquarters being in Boston, Mass.

John Swinscoe, formerly chief engineer of the Driggs-Seabury Ordnance Corporation, Sharon, Pa., manufacturing the Vulcan truck, automobile and truck parts, pressed steel, etc., has entered the employ of the Weston-Mott Co., of Flint, Mich.

George M. Basford, assistant to the president of the American Locomotive Co., New York City, has resigned to become chief engineer of the railroad department of Joseph T. Ryerson & Son, Chicago. Mr. Basford's headquarters will be 30 Church St., New York City.

E. S. Carman, who has given several years to the design and development of various types of molding machines, has entered the employ of the Osborn Mfg. Co., Cleveland, Ohio, as chief engineer in charge of design and construction in the company's molding machine department.

George Bradshaw, general safety agent of the New York Central Lines, has resigned in order to devote his energies to speaking and writing on matters pertaining to safety, and to act in an advisory capacity as a safety engineer. Mr. Bradshaw's home address is Highland, N. Y.

* * *

COMING EVENTS

April 8.—Railroad meeting of the American Society of Mechanical Engineers at 8 P. M., to be held in the Engineering Societies Building, 29 West 39th St., New York City. The main subject for discussion will be "Steel Passenger Car Design."

April 10-12.—Triple joint convention of the National Supply & Machinery Dealers' Association, the Southern Supply & Machinery Dealers' Association and the American Supply & Machinery Manufacturers' Association at Indianapolis, Ind. F. D. Mitchell, secretary-treasurer, 309 Broadway, New York City.

May 15-16.—Semi-annual meeting of the National Machine Tool Builders' Association at the Hotel Astor, New York City. James H. Herron, secretary, 2041 East Third St., Cleveland, Ohio.

May 20-23.—Spring meeting of the American Society of Mechanical Engineers in Baltimore, Md. Hotel Belvedere, headquarters. Layton F. Smith, past president of the Baltimore Engineers' Club, chairman of the local committee. Calvin W. Rice, secretary, 29 W. 39th St., New York City.

May 26-29.—Annual convention of the Master Boiler Makers' Association at Chicago. Harry D. Vought, secretary, 95 Liberty St., New York City.

June 10.—Departure from New York City, of American Society of Mechanical Engineers party to attend joint meeting with Verein deutscher Ingenieure in Leipzig, Germany, beginning June 23.

June 11-13.—Annual convention of the American Railway Master Mechanics' Association at Atlantic City, N. J. Joseph W. Taylor, secretary, Old Colony Bldg., Chicago, Ill.

June 16-18.—Annual convention of the Master Car Builders' Association at Atlantic City, N. J. Joseph W. Taylor, secretary, Old Colony Bldg., Chicago, Ill.

August 16-23.—Second Annual Gas Engine Show of the National Gas Engine Association at Kansas City, Mo. H. R. Brate, secretary, Lakemont, N. Y.

September 17-23.—Third International Congress of Refrigeration to be held in Chicago, Ill. For further information address the secretary-general, Mr. J. F. Nickerson, 431 South Dearborn St., Chicago, Ill.

October 14-16.—Annual convention of the Allied Foundrymen's Associations. Hotel La Salle, headquarters, Richard Moldenke, Watchung, N. J., secretary.

SOCIETIES, SCHOOLS AND COLLEGES

Armour Institute of Technology, Chicago, Ill. Catalogue for 1912-1913.

Massachusetts Institute of Technology, Boston, Mass. Bulletin containing reports of the president and treasurer.

American Museum of Safety, 29 West 39th St., New York City. Year book and bulletin describing the general activities of the museum and the annual meeting and award of medals which took place on January 23. The museum is carrying on a vigorous work for accident prevention, hygiene and mutuality.

American Museum of Safety, 29 West 39th St., New York City. Stories Nos. 1 and 2, entitled "Hans Bright Eyes," "Fleet of Foot" and "Never Touched Me," respectively. These booklets are issued for the purpose of instructing children in the dangers of city streets and how to avoid them. They are particularly suitable for use in schools, especially those of large cities where street traffic is greatly congested.

American Society of Engineer Draftsmen, Walter M. Smyth secretary, 74 Cortlandt St., New York City, has removed its office from 116 Nassau St. to the foregoing address. The program of the regular monthly meeting, March 20, included a paper by Mr. E. S. Houghton, "The Engineer Draftsman's Part on the Erie Barge Canal," and a lecture, "The Application of Descriptive Geometry to Mechanical Drawing," by Mr. Frank Dempster Sherman.

Thaddeus Stevens Industrial School of Pennsylvania, Lancaster, Pa., will hold its first commencement exercises on Friday, March 28. A class of twenty-three students will be graduated, six of whom have taken a course in bricklaying, six in carpentry, five in machine shop practice and six in patternmaking. The instruction of this institution has been arranged to give a thorough knowledge of the theory and practice of each trade, together with instructions in the making and reading of drawings pertaining to the work, particular emphasis being placed on the practical part of the work. The trade courses in this school cover a period of three years.

NEW BOOKS AND PAMPHLETS

Costs of Producing Power in Iowa with Iowa Coals. By H. W. Wagner. 35 pages, 6 by 9 inches. Published by Iowa State College of Agriculture and Mechanic Arts, Ames, Iowa, as Bulletin No. 29.

Data for Use in Designing Culverts and Short-Span Bridges. By Charles H. Moorefield. 39 pages, 6 by 9 inches. Published by the U. S. Department of Agriculture, Washington, D. C., as Bulletin No. 45.

Characteristics and Limitations of the Series Transformer. By A. R. Anderson and H. R. Woodrow. 47 pages, 6 by 9 inches. Published by the University of Illinois Engineering Experiment Station, Urbana, Ill., as Bulletin No. 61.

Report on Incendiarism in Greater New York. By Joseph Johnson, fire commissioner, New York City. 158 pages, 6¾ by 10 inches.

This report, submitted to Mayor Gaynor, contains startling revelations of the extent to which arson has been carried by organized "fire-bugs." The importance of fire losses and insurance makes this report of national interest to property owners.

The Story of Panama. By Frank A. Gause and Charles Carl Carr. 290 pages, 5½ by 8 inches. Illustrated. Published by Silver, Burdett & Co., New York City. Price $1.50.

The near completion of the Panama Canal and the changes likely to be made in the world's commerce by its opening, make the subject of great interest to merchants, mariners, manufacturers, engineers, and, in fact, everyone in the world of trade. This work gives a brief history of the project and a description of the country and many illustrations, showing the canal and conditions of life in the Canal Zone.

Steam Turbines, Their Theory and Construction. By H. Wilda, translated from the German by Charles Sauter. 191 pages, 4½ by 7 inches. Illustrated. Published by Scott, Greenwood & Son, London, England, and D. Van Nostrand Co., New York City. Price $1.25 net.

This book, in condensed form, treats of the principles of steam turbines, the properties of steam, flow of steam, classification of steam turbines according to the arrangement of vanes, flow of steam through the steam turbine, details of steam turbines, general arrangement of various turbines in practice, condensers, utilization of exhaust steam, and practical application of steam turbines.

Essentials of Electricity. By W. H. Timbie. 271 pages, 4¾ by 7 inches. 224 illustrations. Published by John Wiley & Sons, New York City. Price $1.50.

This book is one of the Wiley Technical Series for vocational and industrial schools. The author, who is head of the department of applied science in Wentworth Institute, Boston, Mass., developed the text from notes used in a short trade course for students. It was developed with the idea that a good textbook in any applied subject is one that contains all of the more important facts and principles required by the users. The contents by chapter heads are: "Ohm's Law," "Simple Electric Circuits," "Combinations of Series and Parallel Systems," "Electric Power," "Wire and Wiring Systems," "Generators and Motors," "Locating and Correcting Trouble," "Batteries," "Wiring Diagrams."

Worm Gearing. By Hugh Kerr Thomas. 86 pages, 6 by 9 inches. Published by the McGraw-Hill Book Co., New York City. Price $1.50.

The applications of worm gearing are many, but its use for automobile drive has brought the subject into great prominence during the past few years. This book, which deals exhaustively with the subject, giving an analysis of the principles of worm gearing, should, therefore, be of general value and interest to machine designers and especially those connected with automobile manufacturing plants. The contents by chapter heads are: "Choice of Materials and Methods of Manufacture," "Definitions and Symbols," "Preliminary Proportions," "Pressure Angle and Form of Thread," "Strength of Worm-wheel Teeth," "Stresses in Worm Gearing," "The Width of the Worm-wheel," "The Temperature Coefficient," "Efficiency of Worm Gearing," "General Points of Design of Mounting," "Recapitulation of Formulas Used."

Engineers' Handbook on Patents. By William Macomber. 288 pages, bound in flexible leather, 4½ by 6¾ inches. Published by Little, Brown & Co., Boston, Mass. Price $2.50 net.

The author states in his introduction that the typical American inventor at one time was a clever Yankee who could sit and whittle and whistle, but asserts that the day of the clever Yankee has passed and that the important inventions of the future will be made by men trained in engineering principles. The successful inventor will also be one who understands patent law in its general aspect and who can therefore present his claims intelligently drawn—claims that cover what he wishes to cover and which omit nothing vital to the success of his invention. The contents comprise the general definition of a patent, the nature of invention, what is patentable, patentable novelties, the obtaining of patents, claim construction, infringement, patent litigation, property rights, etc. The book is well printed with large type and is illustrated. Inventors, engineers and others will find the book of interest and value if in any way concerned with invention and the protection of inventions.

Cost Reports for Executives. By Benjamin A. Franklin. 149 pages, 5½ by 8½ inches. Published by the Engineering Magazine Co., New York City. Price $5.

The author discusses the philosophy of costs and the importance of systems which truly record all the items of manufacturing cost. The work is not, however, one that professes to show executives how to construct a cost system, but one that shows him what a cost system should include. The book illustrates with actual forms filled with figures, the points that the writer wishes to make clear. The contents comprise "The Philosophy of Costs," "The Place of the Trial Balance in the Cost System," "The Cost of the Salable Article," "The Economic Consideration of Material by Costs," "Labor from the Cost Viewpoint," "The Vexing Question of Expense," "Statistics as an Aid," "Cost-System—the Basic Improvements." The book is one that should appeal to managerial officers generally, as the author has approached the subject from a new point of view. He seeks to interpret cost totals and to separate those totals into their significant parts. In other words it is the cost system for the executive—not for the cost clerk.

Gas Power. By C. F. Hirshfeld and T. C. Ulbricht. 209 pages, 4¼ by 7½ inches. 60 illustrations. Published by John Wiley & Sons, New York City. Price $1.25 net.

This textbook is intended to give the student such knowledge of the methods and facts of gas power engineering as will enable him to understand its present commercial status. The treatment is as simple and non-mathematical as possible, considering the nature of the subject. The contents by chapter heads are: "The Heat Engine Problem;" "Fuels," "Comparison of the External and Internal Combustion Principles," "Historical Development," "Four- and Two-Stroke Operations," "Methods of Cooling," "Governing and Governors," "Ignition Systems," "Carbureting and Carbureters," "Gas Producers," "Classification and Types of Modern Engines," "Modern Types of Gas and Gasoline Engines," "Modern Types of Oil Engines," "Gas Engine Auxiliaries," "American Practice in the Rating of Internal-Combustion Engines," "Methods of Testing," "Performance of American Engines," "Practical Operations." Comparatively little space is given to the automobile engine, the greater part of the work being devoted to horizontal and vertical stationary engines for power generation.

NEW CATALOGUES AND CIRCULARS

Henry G. Thompson & Son Co., New Haven, Conn. Circular of the "Carr" planer tools.

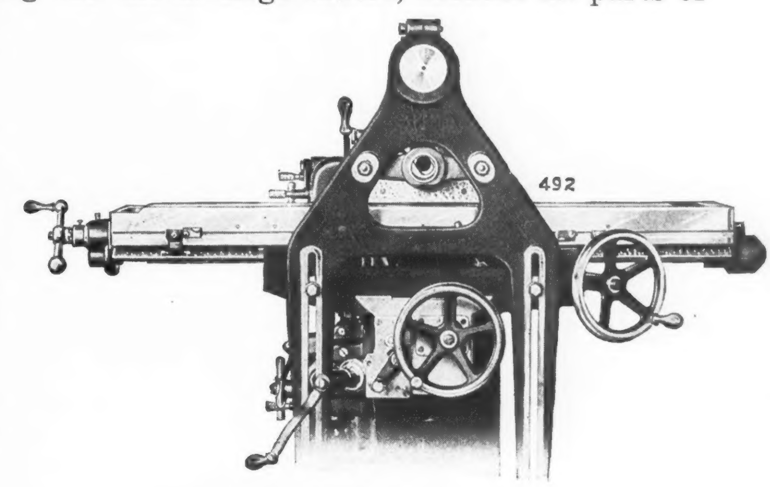
Cincinnati Planer Co., Cincinnati, Ohio. Booklet entitled "A Ragtime Essay on Speed."

Chicago Pneumatic Tool Co., Fisher Bldg., Chicago, Ill. Bulletin No. 130 on lubrication of pneumatic tools, lubricants and automatic oiling devices.

Cincinnati Rigidity

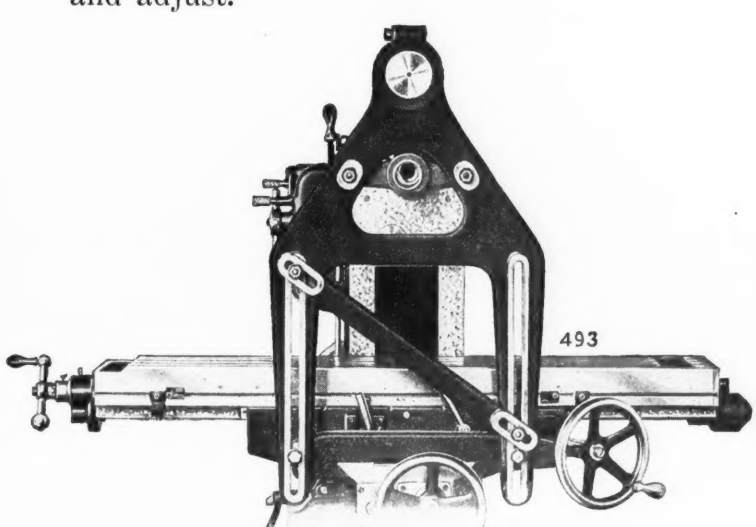
The weakest parts of any Column and Knee Type Miller are the arbor, knee and arbor supports. Their importance is emphasized by the various efforts of designers to strengthen them. The size of arbor is determined by the cutters and is therefore beyond the designer's control. For years we have been recommending the use of large arbors, because all parts of our machines are stronger than the largest arbors in use.

The practical problem resolves itself into providing the greatest rigidity in the other members by making the strongest continuous structure of the Column, Knee, Brace and Overarm.



Usual position of Table when doing heavy cutting

Thorough experiments and careful analyses have shown us that the resultant of the pressures against the outer arbor supports (the braces and overarm) on ordinary work is nearly in a horizontal plane. We designed our braces to resist such pressures, and did not forget to make them, and the arbor bearing, as handy as possible to set up and adjust.



Unusual position of Table
Braces provided with an additional truss member

We employ a single piece triangle. It is always a triangle. Its form is not changed by different positions of the table. In the lower positions the greatest rigidity is needed, and then we add another truss member, making it a double triangle.

The upper ends of the usual two-piece form of triangular braces do not cross when the table is low and they no longer form a triangle at the very time when it is needeed most. Ours, being made in *one* piece, is handy to attach or remove. It is easy to slide it, as well as the arbor supports, along our large cylindrical arm.

This contributes to that handiness in "setting up" that makes our machine more productive.

THE CINCINNATI MILLING MACHINE CO.

CINCINNATI, OHIO, U. S. A.

CONTENTS FOR APRIL

The first column of page numbers applies to the Engineering Edition; the second to the Shop Edition

George P. Clark Co., Windsor Locks, Conn. Bulletin B, illustrating and describing a line of two-wheel, barrel, keg, box and coal hand trucks, steel coal cars, etc.

Wisconsin Miller Mfg. Co., Station A, Milwaukee, Wis. Circular of the Wisconsin hand milling machine built with or without column for floor or bench use.

Bristol Co., Waterbury, Conn. Bulletins No. 138, 139 and 1200 on mechanical time recorders, electric time recorders and Class II recording thermometers, respectively.

Chicago Pneumatic Tool Co., Fisher Bldg., Chicago, Ill. Bulletin No. 126, on Compression Riveters; Bulletin No. 129, on Hose, Hose Couplings and Hose Clamp Tools.

General Electric Co., Schenectady, N. Y. Bulletins No. A 4087 and A 4092 on direct-current motor-starting panels for heavy service and Thomson watt-hour meters, respectively.

Cincinnati Precision Lathe Co., Dept. D, Fosdick Bldg., Cincinnati, Ohio. Card illustrating the "Cincinnati" precision lathe, the headstock construction and the grinding attachment.

Peck-Hamre Mfg. Co., Berlin, Wis. Illustrated circular describing the "Gem" all metal machinists' tool case, which has recently been placed on the market by this company.

American Thermo-Ware Co., 16 Warren St., New York City. Circular of "Autocrat" eye protectors and respirating devices to safeguard the eyes and lungs of grinders, buffers and polishers.

International Oxygen Co., 115 Broadway, New York City. Bulletins 9 and 10 on the I. O. C. system of generating oxygen and hydrogen for industrial purposes, and oxygen and hydrogen generators type 200, respectively.

Triumph Electric Co., Cincinnati, Ohio. Bulletin No. 1010 on adjustable speed motors. The bulletin contains a description of the motors and shows their application to a number of different types of machine tools.

Hess-Bright Mfg. Co., 17 E. Erie Ave., Philadelphia, Pa. Folder showing the construction and giving the dimensions of ball bearing hangers. The folder also calls attention to the power and lubricant savings effected.

Reade Mfg. Co., 1021-1027 Grand St., Hoboken, N. J. Folder advertising the "Incomparable" soldering fluid used for soldering metal work in general. The folder also covers tinning solutions and soldering solutions, giving prices.

Lauth-Juergens Motor Car Co., Fremont, Ohio. Catalogue of 1912 motor trucks, showing in detail some of the more important constructional features of the trucks manufactured by the company, as well as excellent halftone engravings of different types.

Electric Controller & Mfg. Co., Cleveland, Ohio. Bulletin 1031 on type DR drum controllers for direct-current motors. The bulletin contains a price list for 110 volt, 220 volt and 550 volt apparatus, giving also the horsepower ratings.

Garvin Machine Co., Spring and Varick Sts., New York City. Circular No. 192 on the Garvin automatic oil-grooving machine which cuts straight or spiral grooves, right or left hand, in straight or taper holes, in bronze, cast iron, steel, etc.

Challenge Machine Co., Inc., 5116 Springfield Ave., Philadelphia, Pa. Circular of the "Challenge" floor type or "Universal" emery wheel guard, consisting of a standard bolted to the floor and supporting a curved guard partially encircling the wheel.

Armstrong Mfg. Co., 297 Knowlton St., Bridgeport, Conn. Catalogue of Armstrong pipe threading machines and parts, illustrating the machines and all the parts. Each piece is given a number by which it can be identified by users when ordering repair parts.

Bryant Chucking Grinder Co., Springfield, Vt. Circular of the Bryant hole and face grinder for the production of interchangeable work. The machine grinds holes up to 10 inches in diameter, 6 inches in length, and has a chuck range for work up to 12 inches in diameter.

Eugene Dietzgen Co., 218 E. 23d St., New York City. Circular of drafting-room furniture, comprising folding drawing stands, adjustable drawing tables, wooden horses, drawing boards, etc.; also circular of sectional filing cases and combination filing cabinets for drawing-rooms.

Warner & Swasey Co., Cleveland, Ohio. General catalogue No. 15 on hollow hexagon turret lathes, universal hollow turret lathes, turret screw machines, turret lathes, set-over turret lathes, forming turret lathes, miscellaneous brass working machine tools, tools and accessories, etc.

Ingersoll-Rand Co., 11 Broadway, New York City. Circular Form No. 3008 on class "PE" duplex direct-connected electrically-driven air compressors; Form No. 575, instructions for installing and operating class "PE" direct-connected electric driven air compressors with duplicate part list.

Hess-Bright Mfg. Co., 17 E. Erie Ave. Philadelphia, Pa. Data sheets on ball bearings and their correct uses, comprising thrust collar bearings, light weight series No. 1000; thrust collar bearings, medium weight series No. 1100; and ball bearing mountings for worm-driven capstan.

Monash-Younker Co., 121 W. 42nd St., New York City. Circular advertising the Monash noiseless system of vacuum heating, using exhaust steam for heating and drying purposes. The circular contains a list of plants equipped with this system of heating and describes the apparatus.

Baldwin Chain & Mfg. Co., Worcester, Mass. Catalogue of transmission chains and sprockets. The catalogue contains, in addition to tables of dimensions of chains and sprockets, formulas and directions for calculating the length of chains, and notes on the proper design of sprockets.

Thwing Instrument Co., 445 North Fifth St., Philadelphia, Pa. Bulletin V on Type A and Type C recording pyrometers. An important feature of the Thwing pyrometers is the multiple record system from which two or three records may be made from as many heat sources by one galvanometer upon one chart or section of chart.

Manning, Maxwell & Moore, 85-89 Liberty St., New York City. Catalogue F, issued by the Ashcroft Mfg. Co. (a constituent concern) on steam, pressure and vacuum gages, Edison recording gages, Tabor engine indicators, engineering specialties, pipe tools, clocks, revolution counters, test pumps, low water detectors, reducing wheels, pantographs, planimeters, etc.

General Electric Co., Schenectady, N. Y. Bulletins No. A 4070, Electrically-Operated Remote Control Switch; A 4072 Sign Lighting by Edison Mazda Sign Lamps; A 4081, Reversing Motors for Planers, Slotters, etc.; A 4083, Roller Bearing Trolley Bases; A 4085 Battery-charging Motor-Generator Sets for Railway Signalling; A 4086, Circuit Breakers for Railway Service.

Heine Safety Boiler Co., St. Louis, Mo. Pamphlet on superheating, containing a reprint of the paper from the Engineers' Society of Pennsylvania, discussing the steam and fuel economy of superheating, and a brief description of the design and operation of the Heine superheater; also abstracts from the Transactions of the A. S. M. E. on the effect of superheated steam on cast iron.

Canton Foundry & Machine Co., Canton, Ohio. Catalogue of the Canton portable floor crane and hoist, with which one man can hoist and convey to any part of the shop, a piece weighing up to 4000 pounds and deposit it on a lathe, planer, truck bed, etc. The convenience of this apparatus is illustrated by means of views showing uses in machine shops, roundhouses, foundries, etc.

Landis Machine Co., Inc., Waynesboro, Pa. 1913 catalogue, No. 21, covering 79, 6 by 9 inch pages illustrating and describing bolt, pipe and nipple threading machinery, bolt pointing and nut tapping machinery, screw cutting die-heads and special threading machines. The catalogue contains interesting information on the action of die chasers and shows in detail the construction of chaser holders, rotary die-heads, tap chucks, bolt pointer heads, reaming devices and nipple grips.

Sprague Electric Works of General Electric Co., 527 W. 34th St., New York City. Bulletin No. 115, on Electrical Equipment for Buildings, including generators, generating sets, switchboards, wire and cable, panel boards, motors, controllers, electric hoists and fans, etc.; Catalogue No. 327, on Sprague Electric Fans, giving dimensions, weights and prices, and illustrating the different types of fans made; Bulletin No. 241, on Motor Drive for Stereotyping and Electrotyping Machinery, and Electroplating Generators.

Titanium Alloy Mfg. Co., Niagara Falls, N. Y. Booklet entitled "Titanium in Steel" containing interesting statements from steel makers and scientists on the uses and benefits of ferro carbon-titanium. The claim made for this deoxidizer is that when used for scavenging purposes it not only thoroughly cleanses the metal but leaves none of its compounds in the finished product. The booklet, which contains 48 pages, 4¼ by 8½ inches, is an addition to metallurgical literature, and should be appreciated by those concerned with the use of high-grade steels.

TRADE NOTES

Windsor Machine Co., Windsor, Vt., is converting its old shop into a club-house for the use of its employes.

Jones & Lamson Machine Co., Springfield, Vt., is building an addition to its lower plant 110 by 400 feet. The addition is of sawtooth construction in harmony with the construction of the present plant.

Western Pump & Engineering Co., 339-343 Railway Exchange, Chicago, Ill., is bringing out a line of triplex and centrifugal pumps and air compressors in addition to its line of single and duplex steam pumps.

Hendee Mfg. Co., Springfield, Mass., has made plans to double the capacity of its plant for manufacturing motor-cycles. The production for the 1913